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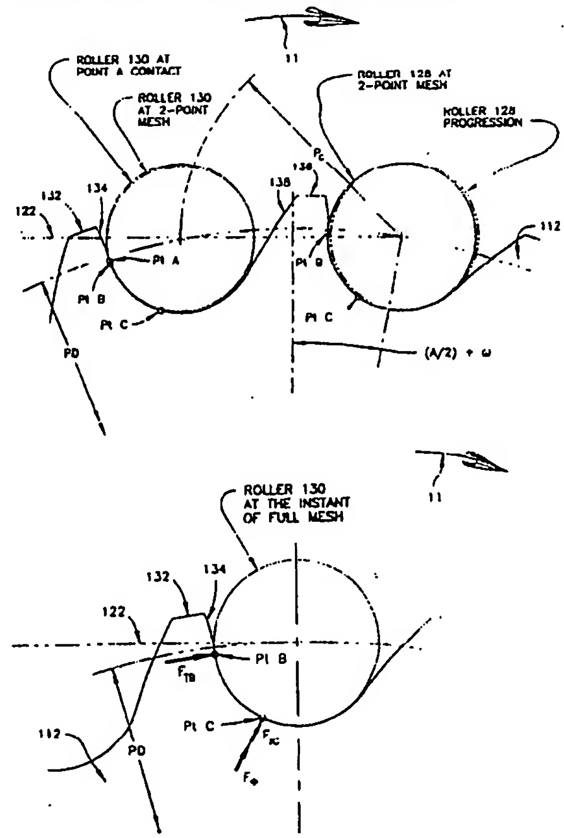
(54) Title: RANDOM ENGAGEMENT ROLLER CHAIN SPROCKET HAVING IMPROVED NOISE CHARACTERISTICS

(57) Abstract

A unidirectional roller chain sprocket (112) for use primarily in automotive engine drive chain applications which incorporates an asymmetrical tooth for improved noise reduction. The sprocket (112) includes a first plurality of sprocket teeth (132) each having a first engaging flank (134) with a first contact point at which a roller (128) contacts the first engaging flank (134), and a second plurality of sprocket teeth (136) each having a second engaging flank with a second contact point at which a roller (128) contacts the second engaging flank. The first engaging flanks (134) include a flank flat (144) which facilitates spacing the first contact point on the first engaging flank (134) relative to the second contact point on the second engaging flank to effect a time delay between an initial roller to first sprocket tooth contact and an initial roller to second sprocket tooth contact. The flank flat (144) is tangent to an engaging flank radius Rf and a first root radius. The asymmetrical tooth profile may also incorporate one or more inclined root surfaces which provide tooth space clearance for maintaining the chain rollers in hard contact with the root surface in the sprocket wrap, or added pitch mismatch wherein the sprocket chordal pitch is less than the chain link pitch to facilitate a "staged" roller-tooth contact as a roller (128) moves into full mesh from an initial tangential impact at the flank flat (144).



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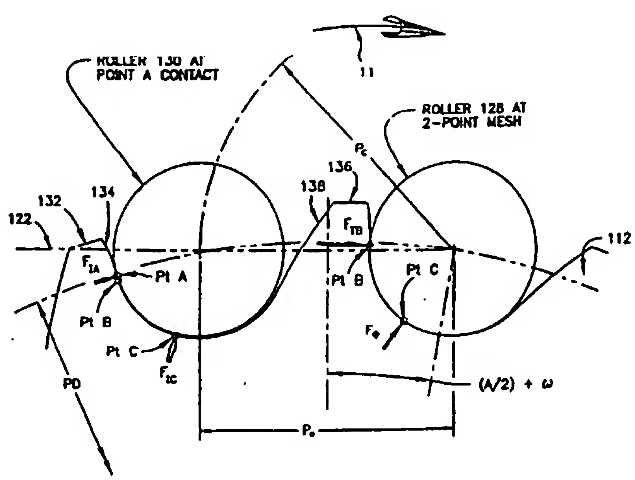

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RANDOM ENGAGEMENT ROLLER CHAIN SPROCKET HAVING IMPROVED NOISE CHARACTERISTICS

Cross Reference to Related Applications

This application claims the benefit of U.S. Provisional Patent Application Nos. 60/022,321, filed July 25, 1996 and 60/032,379, filed December 19, 1996.

Background of the Invention

The present invention relates to the automotive timing chain art. It finds particular application in conjunction with a unidirectional roller chain sprocket for use in automotive camshaft drive applications and will be described with particular reference thereto. However, the present invention may also find application in conjunction with other types of chain drive systems and applications where reducing the noise levels associated with chain drives is desired.

Roller chain sprockets for use in camshaft drives of automotive engines are typically manufactured according to ISO (International Organization for Standardization) standard 606:1994(E). The ISO-606 standard specifies requirements for short-pitch precision roller chains and associated chain wheels or sprockets.

Figure 1 illustrates a symmetrical tooth space form for an ISO-606 compliant sprocket. The tooth space has a continuous fillet or root radius R_i from one tooth flank (i.e., side) to the adjacent tooth flank. A chain with a link pitch P has rollers of diameter D_1 in contact with the tooth spaces. The ISO sprocket has a chordal

pitch also of length P, a root diameter D_2, and Z number of teeth. The tooth flank radius R_f, pitch circle diameter PD, tip diameter OD, tooth angle A (equal to 360°/Z), and roller seating angle α further define the ISO-606 compliant sprocket. The maximum and minimum roller seating angle α is defined as:

$$\alpha_{max} = (140° - 90°)/Z \quad \text{and} \quad \alpha_{min} = (120° - 90°)/Z$$

With reference to Figure 2, an exemplary ISO-606 compliant roller chain drive system 10 rotates in a clockwise direction as shown by arrow 11. The chain drive system 10 includes a drive sprocket 12, a driven sprocket 14 and a roller chain 16 having a number of rollers 18. The sprockets 12, 14, and chain 16 each generally comply with the ISO-606 standard.

The roller chain 16 engages and wraps about sprockets 12 and 14 and has two spans extending between the sprockets, slack strand 20 and taut strand 22. The roller chain 16 is under tension as shown by arrows 24. The taut strand 22 may be guided from the driven sprocket 14 to the drive sprocket 12 with a chain guide 26. A first roller 28 is shown at the onset of meshing at a 12 o'clock position on the drive sprocket 12. A second roller 30 is adjacent to the first roller 28 and is the next roller to mesh with the drive sprocket 12.

Chain drive systems have several components of undesirable noise. A major source of roller chain drive noise is the sound generated as a roller leaves the span and collides with the sprocket during meshing. The resultant impact noise is repeated with a frequency generally equal to that of the frequency of the chain meshing with the sprocket. The loudness of the impact noise is a function of the impact energy (E_A) that must be absorbed during the meshing process. The impact energy absorbed is related to engine speed, chain mass, and the impact velocity between the chain and the

sprocket at the onset of meshing. The impact velocity is affected by the chain-sprocket engagement geometry, of which an engaging flank pressure angle γ is a factor, where:

$$E_A = \frac{wP}{2000} V_A^2 ;$$

$$V_A = \frac{\pi n P}{30000} \sin\left(\frac{360}{Z} + \gamma\right);$$

$$\gamma = \frac{180 - A - \alpha}{2}; \text{ and}$$

E_A	=	Impact Energy [N·m]
V_A	=	Roller Impact Velocity [m/s]
γ	=	Engaging Flank Pressure Angle
n	=	Engine Speed [RPM]
w	=	Chain Mass [Kg]
Z	=	Number of Sprocket Teeth
A	=	Tooth Angle (360°/Z)
α	=	Roller Seating Angle
P	=	Chain Pitch (Chordal Pitch)

The impact energy equation presumes the chain drive kinematics will conform generally to a quasi-static analytical model and that the roller-sprocket driving contact will occur at a tangency point TP (Fig. 3) for the flank and root radii as the sprocket collects a roller from the span.

As shown in Figure 3, the pressure angle γ is defined as the angle between a line A extending from the center of the engaging roller 30 normal to and through the tangency point TP of engaging flank radius R_f and root radius R_i and a line B connecting the centers of an imaginary roller fully seated on the theoretical pitch diameter PD, and the engaging roller 30 when fully seated on the theoretical pitch diameter PD. The roller seating angles α and pressure angles γ listed in Figure 27 are calculated from the equations defined above. It

should be appreciated that γ is a minimum when α is a maximum. The exemplary 18-tooth ISO compliant sprocket 12 of Figure 3 will have a pressure angle γ in the range of 12.5° to 22.5° as shown in Figure 27.

Figure 3 also shows the engagement path (phantom rollers) and the meshing contact position of roller 30 (solid) as the drive sprocket 12 rotates in the direction of arrow 11. It is assumed that the chain drive kinematics generally conform to, and can be represented by a quasi-static analytical model, and that the drive contact occurs at the tangency point TP. Figure 3 depicts the theoretical case with chain roller 28 seated on root diameter D_2 of a maximum material sprocket with both chain pitch and sprocket chordal pitch equal to theoretical pitch P. For this theoretical case, the noise occurring at the onset of roller engagement has a radial component F_{Ir} as a result of roller 30 colliding with the root surface R_i and a tangential component F_{It} generated as the same roller 30 collides with the engaging tooth flank at point TP as the roller moves into driving contact. It is believed that the radial impact occurs first, with the tangential impact following nearly simultaneously.

Under actual conditions as a result of feature dimensional tolerances, there will normally be a pitch mismatch between the chain and sprocket, with increased mismatch as the components wear in use. This pitch mismatch serves to move the point of meshing impact, with the radial collision still occurring at the root surface R_i but not necessarily at D_2. The tangential collision will normally be in the proximity of point TP, but this contact could take place high up on the engaging side of root radius R_i or even radially outward from point TP on the engaging flank radius R_f as a function of the actual chain-sprocket pitch mismatch.

Reducing the engaging flank pressure angle γ reduces the meshing noise levels associated with roller

chain drives, as predicted by the impact energy (E_A) equation set forth above. It is feasible but not recommended to reduce the pressure angle γ while maintaining a symmetrical tooth profile, which could be accomplished by simply increasing the roller seating angle α, effectively decreasing the pressure angle for both flanks. This profile as described requires that a worn chain would, as the roller travels around a sprocket wrap (discussed below), interface with a much steeper incline and the rollers would necessarily ride higher up on the coast flank prior to leaving the wrap.

Another source of chain drive noise is the broadband mechanical noise generated in part by shaft torsional vibrations and slight dimensional inaccuracies between the chain and the sprockets. Contributing to a greater extent to the broadband mechanical noise level is the intermittent or vibrating contact that occurs between a worn roller chain and sprocket as the rollers travel around the sprocket wrap. In particular, ordinary chain drive system wear comprises sprocket tooth face wear and chain wear. The chain wear is caused by bearing wear in the chain joints and can be characterized as pitch elongation. It is believed that a worn chain meshing with an ISO standard sprocket will have only one roller in driving contact and loaded at a maximum loading condition.

With reference again to Figure 2, driving contact at maximum loading occurs as a roller enters a drive sprocket wrap 32 at engagement. Engaging roller 28 is shown in driving contact and loaded at a maximum loading condition The loading on roller 28 is primarily meshing impact loading and the chain tension loading. The next several rollers in the wrap 32 forward of roller 28 share in the chain tension loading, but at a progressively decreasing rate. The loading of roller 28 (and to a lesser extent for the next several rollers in

the wrap) serves to maintain the roller in solid or hard contact with the sprocket root surface 34.

A roller 36 is the last roller in the drive sprocket wrap 32 prior to entering the slack strand 20. Roller 36 is also in hard contact with drive sprocket 12, but at some point higher up (e.g., radially outwardly) on the root surface 34. With the exception of rollers 28 and 36, and the several rollers forward of roller 28 that share the chain tension loading, the remaining rollers in the drive sprocket wrap 32 are not in hard contact with the sprocket root surface 34, and are therefore be free to vibrate against the sprocket root surfaces as they travel around the wrap, thereby contributing to the generation of unwanted broadband mechanical noise.

A roller 38 is the last roller in a sprocket wrap 40 of the driven sprocket 14 before entering the taut strand 22. The roller 38 is in driving contact with the sprocket 14. As with the roller 36 in the drive sprocket wrap 32, a roller 42 in the sprocket wrap 40 is in hard contact with a root radius 44 of driven sprocket 14, but generally not at the root diameter.

It is known that providing pitch line clearance (PLC) between sprocket teeth promotes hard contact between the chain rollers and sprocket in the sprocket wrap as the roller chain wears. The amount of pitch line clearance added to the tooth space defines a length of a short arc that is centered in the tooth space and forms a segment of the root diameter D_2. The root fillet radius R_i is tangent to the flank radius R_F and the root diameter arc segment. The tooth profile is still symmetrical, but R_i is no longer a continuous fillet radius from one flank radius to the adjacent flank radius. This has the effect of reducing the broadband mechanical noise component of a chain drive system. However, adding pitch line clearance between

sprocket teeth does not reduce chain drive noise caused by the roller-sprocket collision at impact.

Chordal action, or chordal rise and fall, is another important factor affecting the operating smoothness and noise levels of a chain drive, particularly at high speeds. Chordal action occurs as the chain enters the sprocket from the free span during meshing and it can cause a movement of the free chain in a direction perpendicular to the chain travel but in the same plane as the chain and sprockets. This chain motion resulting from chordal action will contribute an objectionable noise level component to the meshing noise levels, so it is therefore beneficial to reduce chordal action inherent in a roller chain drive.

Figure 4 illustrates the chordal action for an 18-tooth ISO compliant sprocket having a chordal pitch of 9.525 mm. Chordal rise may conventionally be defined as the displacement of the chain centerline as the sprocket rotates through an angle A/2, where:

$$\text{Chordal rise} = r_p - r_c = r_p[1-\cos(180°/Z]$$

where r_c is the chordal radius, or the distance from the sprocket center to a pitch chord of length P; r_p is the actual theoretical pitch radius; and Z is the number of sprocket teeth.

It is known that a short pitch chain provides reduced chordal action compared to a longer pitch chain having a similar pitch radius. Figure 4 assumes a driven sprocket (not shown) also having 18-teeth and in phase with the drive sprocket shown. In other words, at T = 0, both sprockets will have a tooth center at the 12 o'clock position. Accordingly, this chain drive arrangement under quasi-static conditions will have a top or taut strand that will move up and down in a uniform manner a distance equal to that of the chordal rise. At T = 0, roller 46 is at the onset of meshing,

with chordal pitch P horizontal and in line with taut strand 22. At T = 0 + (A/2), roller 46 has moved to the 12 o'clock position.

For many chain drives, the drive and driven sprockets will be of different sizes and will not necessarily be in phase. The chain guide 26 (Fig. 2) has the primary purpose to control chain strand vibration in the free span. The geometry of the guide-chain interface also defines the length of free span chain over which chordal rise and fall is allowed to articulate. Figure 5 is an enlarged view of Figure 2 showing the first roller 28 at the onset of engagement and a second roller 30 as the next roller about to mesh with sprocket 12. In this example, the chain guide 26 controls and guides the taut strand 22 except for five (5) unsupported link pitches extending between the chain guide 26 and the engaging roller 28. The taut strand 22 is horizontal when the roller 28 is at the 12 o'clock position.

With reference to Figures 6 and 7, the drive sprocket 12 is rotated in a clockwise direction to advance roller 28 to a new angular position (A/2) + ω, determined by the instant of sprocket engagement of roller 30. Roller 28 is considered to be seated and in hard contact with the root surface at D_2 at the onset of meshing of roller 30. As shown in Figure 6, a straight line is assumed for the chain span from roller 28 to a chain pin center 48, about which the unsupported span from pin 48 to engaging roller 30 is considered to rotate. The location and chain-interfacing contour of the chain guide 26 will determine the number of free span pitches about which articulation will take place as a result of the chordal rise and fall during the roller meshing process.

As best seen in Figure 7, assuming that rollers 28 and 30 are in hard contact with the sprocket root surfaces at D_2, the chordal rise is the

perpendicular displacement of the center of roller 30 (located on the pitch diameter PD) from the taut span 22 path as it moves from its initial meshing position shown to the position presently occupied by roller 28.

Accordingly, it is desirable to develop a new and improved roller chain drive system which meets the above-stated needs and overcomes the foregoing disadvantages and others while providing better and more advantageous results.

Summary of the Invention

In accordance with one aspect of the present invention, a roller chain sprocket is disclosed. The roller chain sprocket includes a first plurality of sprocket teeth each having a first tooth profile including a first engaging flank with a first contact point at which a roller initially contacts the first engaging flank, a second plurality of sprocket teeth each having a second tooth profile including a second engaging flank with a second contact point at which a roller initially contacts the second engaging flank; and the first tooth profile including a flank flat on the first engaging flank which facilitates spacing the first contact point on the first engaging flank relative to the second contact point on the second engaging flank to alter an initial roller-to-first engaging flank contact relative to an initial roller-to-second engaging flank contact. Thus, the flank flat facilitates altering the meshing contact from the first profile to the second. That is, while the frequency of the contacts (at constant rpm) will be substantially identical for each tooth profile, the profile differences serve to shift the initial contact phasing, thereby altering the rhythm of the initial contacts from one profile to the other.

In accordance with another aspect of the present invention, a unidirectional roller chain drive system is disclosed. The unidirectional roller chain

drive system includes a driving sprocket having sprocket teeth with engaging flanks and coast flanks, the engaging flanks cooperating with the coast flanks of adjacent teeth to define asymmetrical tooth spaces between the sprocket teeth, a driven sprocket having sprocket teeth with engaging flanks and coast flanks, the engaging flanks cooperating with the coast flanks of adjacent teeth to define asymmetrical tooth spaces between the sprocket teeth, a roller chain having rollers in engaging contact with the driving sprocket and the driven sprocket, and at least one of the driving sprocket and the driven sprocket including a first plurality of sprocket teeth each having a first engaging flank with a first contact point at which a roller initially contacts the first engaging flank, a second plurality of sprocket teeth each having a second engaging flank with a second contact point at which a roller initially contacts the second engaging flank, and the first engaging flank including a flank flat which facilitates phasing a frequency of initial roller-to-first engaging flank contacts relative to initial roller-to-second engaging flank contacts to alter the rhythm of the initial roller-to-first engaging flank and the roller-to-second engaging flank contacts. As with the first aspect of the invention, the flank flat facilitates altering the meshing contact from the first profile to the second.

In accordance with yet another aspect of the present invention, a roller chain sprocket is disclosed. The roller chain sprocket includes a first plurality of sprocket teeth each having a first engaging flank with a first contact point at which a roller initially contacts the first engaging flank, a second plurality of sprocket teeth each having a second engaging flank with a second contact point at which a roller initially contacts the second engaging flank, and the first engaging flank including a flank flat which facilitates

phasing a frequency of initial roller-to-first engaging flank contacts relative to initial roller-to-second engaging flank contacts to alter the rhythm of the initial roller-to-first engaging flank and the roller-to-second engaging flank contacts.

One advantage of the present invention is the provision of a roller chain sprocket which incorporates a flank flat on an engaging tooth surface which facilitates altering a meshing contact from a first tooth profile to a second tooth profile.

Another advantage of the present invention is the provision of a roller chain sprocket which incorporates a flank flat on an engaging tooth surface which effects a time delay between an initial roller-to first sprocket tooth profile contact and an initial roller-to-second sprocket tooth profile contact.

Another advantage of the present invention is the provision of a roller chain sprocket which incorporates a flank flat on an engaging tooth surface of a first tooth profile which facilitates phasing a frequency of initial roller-to-engaging flank contacts of the first tooth profile relative to initial roller-to-engaging flank contacts of a second tooth profile to alter the rhythm of the initial roller-to-first engaging flank and the roller-to-second engaging flank contacts.

Another advantage of the present invention is the provision of a roller chain sprocket which incorporates added pitch mismatch between the sprocket and roller chain to facilitate a "staged" roller-to-sprocket impact.

Still another advantage of the present invention is the provision of a roller chain sprocket which incorporates an inclined root surface on an engaging flank, a coast flank, or both an engaging flank and a coast flank to provide tooth space clearance.

Yet another advantage of the present invention is the provision of a roller chain sprocket which

minimizes impact noise generated by a roller-sprocket collision during meshing.

A further advantage of the present invention is the provision of a roller chain sprocket which minimizes broadband mechanical noise generated by unloaded rollers in a sprocket wrap.

A still further advantage of the present invention is the provision of a roller chain sprocket which provides a "staged" roller impact wherein a tangential impact occurs first followed by a two-point radial impact.

Yet a further advantage of the present invention is the provision of a roller chain sprocket which spreads roller engagement over a significant time interval to provide for a more gradual load transfer, thereby minimizing roller-sprocket impact and the inherent noise generated therefrom.

Yet a further advantage of the present invention is the provision of a roller chain sprocket having two sets of sprocket teeth incorporating different tooth profiles which cooperate to reduce chain drive system noise levels below a noise level which either tooth profile used alone would produce.

Further advantages of the present invention will become apparent to those of ordinary skill in the art upon reading and understanding the following detailed description of the preferred embodiments.

Brief Description of the Drawings

The invention may take form in various components and arrangements of components, and in various steps and arrangements of steps. The drawings are only for purposes of illustrating a preferred embodiment and are not to be construed as limiting the invention.

Figure 1 illustrates a symmetrical tooth space form for a ISO-606 compliant roller chain sprocket;

Figure 2 is an exemplary roller chain drive system having a ISO-606 compliant drive sprocket, driven sprocket, and roller chain;

Figure 3 shows an engagement path (phantom) and an initial contact position (solid) of a roller as a ISO-606 compliant drive sprocket rotates in a clockwise direction;

Figure 4 shows a roller at the onset of meshing with the drive sprocket of Figure 2, and shows the drive sprocket rotated in a clockwise direction until the roller is at a 12 o'clock position;

Figure 5 is an enlarged view of the drive sprocket of Figure 2 with a roller fully seated in a tooth space and a second roller about to mesh with the drive sprocket;

Figure 6 shows the drive sprocket of Figure 5 rotated in a clockwise direction until the second roller initially contacts the drive sprocket;

Figure 7 is an enlarged view of Figure 6 showing that the second roller initially contacts a root surface (i.e., radial impact) of the drive sprocket, under theoretical conditions;

Figure 8 illustrates a roller chain drive system having a roller chain drive sprocket and driven sprocket which incorporate the features of the present invention therein;

Figure 9 illustrates the roller chain drive sprocket of Figure 8 with an asymmetrical tooth space form in accordance with a first embodiment of the present invention;

Figure 10 is an enlarged view of the asymmetrical tooth space form of Figure 9 showing a roller in two-point contact with the sprocket;

Figure 11 shows an engagement path (phantom) and an initial contact position (solid) of a roller as the drive sprocket of Figure 8 rotates in a clockwise direction;

Figure 12 is an enlarged view of the drive sprocket of Figure 8 with a roller fully seated in a tooth space and a second roller as the next roller to be collected from a taut span of the roller chain;

Figure 13 shows the drive sprocket of Figure 12 rotated in a clockwise direction until the second roller initially contacts the drive sprocket;

Figure 14 is an enlarged view of Figure 13 showing the first roller in two-point contact and second roller at initial tangential contact with the drive sprocket;

Figure 14a illustrates the progression of a first and a second roller as the roller chain drive sprocket of Figure 8 is rotated in a clockwise direction;

Figure 14b is an enlarged view of the drive sprocket of Figure 14 rotated in a clockwise direction to advance the second roller to the instant of full mesh at a 12 o'clock position;

Figure 15 illustrates a roller chain drive sprocket with an asymmetrical tooth space form in accordance with a second embodiment of the present invention;

Figure 16 is an enlarged view of Figure 8, showing the contact progression as the rollers travel around the drive sprocket wrap;

Figure 17 is an enlarged view of a roller exiting a sprocket wrap of the sprocket of Figure 8;

Figure 18 illustrates a roller chain sprocket with an asymmetrical tooth space form in accordance with a third embodiment of the present invention;

Figure 19 illustrates a roller chain sprocket with an asymmetrical tooth space form in accordance with a fourth embodiment of the present invention;

Figure 20 is a side view of an exemplary random-engagement roller chain sprocket which

incorporates the features of the present invention therein;

Figure 21 illustrates the random-engagement roller chain sprocket of Figure 20 with an additional asymmetrical tooth space form in accordance with the present invention;

Figure 22 illustrates the asymmetrical tooth space form of Figure 9 overlaid with one embodiment of the asymmetrical tooth space form of Figure 21;

Figure 23 illustrates the asymmetrical tooth space form of Figure 9 overlayed with a second embodiment of the asymmetrical tooth space form of Figure 21;

Figure 24 illustrates the progression of a first and a second roller as the random-engagement roller chain sprocket of Figure 20 is rotated in a clockwise direction;

Figure 25 illustrates the sprocket of Figure 23 with a first roller in two-point contact, a second roller at initial tangential contact, and a third roller about to mesh with the drive sprocket;

Figure 26 illustrates the sprocket of Figure 25 rotated in a clockwise direction until the instant of initial tangential contact between the third roller and the roller chain drive sprocket;

Figure 27 is a table listing roller seating angles α and pressure angles γ for a number of different ISO-complaint sprocket sizes; and

Figure 28 is a table listing the maximum Beta (β) angles and the corresponding pressure angles (γ) for a number of different ISO-compliant and several asymmetrical sprocket sizes.

<u>Detailed Description of the Preferred Embodiments</u>

With reference now to Figure 8, a roller chain drive system 110 includes a drive sprocket 112 and a driven sprocket 114 which incorporate the features of

the present invention therein. The roller chain drive system 110 further includes a roller chain 116 having a number of rollers 118 which engage and wrap about sprockets 112, 114. The roller chain rotates in a clockwise direction as shown by arrow 111.

The roller chain 116 has two spans extending between the sprockets, slack strand 120 and taut strand 122. The roller chain 116 is under tension as shown by arrows 124. A central portion of the taut strand 122 may be guided from the driven sprocket 114 to the drive sprocket 112 with a chain guide 126. A first roller 128 is shown fully seated at a 12 o'clock position on the drive sprocket 112. A second roller 130 is adjacent to the first roller 128 and is about to mesh with the drive sprocket 112.

To facilitate the description of an asymmetrical tooth profile of the present invention, reference will be made only to the drive sprocket 112. However, the asymmetrical tooth profile of the present invention is equally applicable to the driven sprocket 114, as well as to idler sprockets and sprockets associated with counter rotating balance shafts.

Referring now to Figures 9 and 10, the sprocket 112 includes a first tooth 132 having an engaging flank 134, and a second tooth 136 having a coast or disengaging flank 138. The engaging flank 134 and coast flank 138 cooperate to define a tooth space 140 which receives the engaging roller 128 (shown in phantom). The engaging roller 128 has a roller diameter D_1, and is shown fully seated in two-point contact within the tooth space 140 as described further below. More particularly, the engaging roller 128, when fully seated in the tooth space, contacts two lines B and C that extend axially along each sprocket tooth surface (i.e.., in a direction orthogonal to the plane of the drawings). However, to facilitate a description thereof, the lines

A, B, and C are hereafter shown and referred to as contact points within the tooth space.

The engaging flank 134 has a radius R_f which is tangent to a radially outer end of a flank flat 144. The location of the flank flat 144 is defined by an angle β, with the flat orientation being normal or perpendicular to a line that passes through Point B and the center of roller 128 when the roller is contacting the sprocket at Points B and C. The length of the flank flat extending radially outward from Point B affects a time delay between an initial tangential impact between sprocket 112 and roller 128 at a first contact Point A along the flank flat 144, and a subsequent radial impact at Point C. It is believed that the roller stays in contact with the flank flat from its initial tangential contact at Point A until the roller moves to a fully engaged two-point contact position at Points B and C. The pressure angle γ, the amount of pitch mismatch between the chain and the sprocket, and the length of the flank flat can be varied to achieve a desired initial roller contact Point A at the onset of roller-sprocket meshing.

It should be appreciated that flank (tangential) contact always occurs first, with radial contact then occurring always at Point C regardless of chain pitch length. In contrast, with known tooth space forms (e.g., ISO compliant and asymmetrical) incorporating single point contact (e.g. single line contact), an engaging roller must move to a driving position after making radial contact. The pressure angles γ therefore assume that the engaging roller will contact at the flank radius/root radius tangency point. Thus, the meshing contact location of the known single point/line tooth space forms is pitch "sensitive" to determine where the radial impact as well as tangential impact will occur.

The engaging flank roller seating angle β (Fig. 9) and a disengaging flank roller seating angle β' replace the ISO-606 roller seating angle α (ISO profile shown in phantom). The pressure angle γ is a function of the engaging flank roller seating angle β. That is, as β increases, γ decreases. A minimum asymmetrical pressure angle can be determined from the following equation, where:

$$\gamma_{min} = \beta_{max} - (\alpha_{max} /2 + \gamma_{ISO\ min})$$

Therefore, an asymmetrical pressure angle $\gamma_{min} = 0$ when $\beta_{max} = (\alpha_{max} /2 + \gamma_{ISO\ min})$ as illustrated in Figure 28. Figure 28 lists the maximum Beta (β) angles and the corresponding pressure angles (γ) for several sprocket sizes and several asymmetrical profiles. It should be appreciated that reducing the engaging flank pressure angle γ reduces the tangential impact force component F_{IA} (Fig. 14) and thus the tangential impact noise contribution to the overall noise level at the onset of engagement.

Impact force F_{IA} is a function of the impact velocity which in turn is related to pressure angle γ. As pressure angle γ is reduced, it provides a corresponding reduction in the impact velocity between the chain and the sprocket at the onset of meshing. A minimum pressure angle γ also facilitates a greater separation or distance between tangential contact points A and B to further increase or maximize engagement "staging". In the preferred embodiment, the engaging flank pressure angle γ is in the range of about -2.0° to about +5° to optimize the staged impact between the roller and the sprocket.

In the embodiment being described, roller seating angle β is greater than ISO $\alpha_{max}/2$ at a maximum material condition and β can be adjusted until a desired engaging flank pressure angle γ is achieved. For

instance, the roller seating angle β of Figure 9 provides a pressure angle γ that is less than zero, or a negative value. The negative pressure angle γ is best seen in Figure 11, as contrasted with the ISO-606 compliant tooth profile of Figure 3 with a positive pressure angle γ. It is believed that a small negative pressure angle for the theoretical chain/sprocket interface beneficially provides a pressure angle γ closer to zero (0) for a "nominal" system or for a system with wear. However, the engaging flank roller seating angle β may be beneficially adjusted so as to provide any engaging flank pressure angle γ having a value less than the minimum ISO-606 pressure angle.

Referring again to Figures 9 and 10, a first root radius R_i is tangent to a radially inner end of the flank flat **144**, and tangent to a radially outer end of an inclined root surface **146**. As best seen in Figure 10, a maximum root radius R_i must be equal to, or less than, a minimum roller radius $0.5D_1$ to facilitate the fully engaged two-point/line contact at Points B and C. Accordingly, this will define a small clearance between R_i at the engaging flank **134** and the roller **128** at the two-point/line engagement. The flank flat **144** and the inclined root surface **146** necessarily extend inside Points B and C respectively to facilitate the two-point/line roller contact at engagement as well as the required clearance R_i to the roller **128**. A second root radius R_i' is tangent to a radially inner end of the inclined root surface **146** at line **150**. The coast flank has a radius R_f' with its arc center defined by disengaging side roller seating angle β'.

The inclined root surface **146** is a flat surface having a finite length which defines a tooth space clearance (TSC). The tooth space clearance compensates for chain pitch elongation or chain wear by accommodating a specified degree of chain pitch elongation ΔP. In other words, the tooth space

clearance TSC enables rollers of a worn chain to be maintained in hard contact with the inclined root surface of the sprocket teeth. In addition, the inclined root surface 146 facilitates reducing the radial reaction force thereby reducing the roller radial impact noise contribution to the overall noise level.

The inclined root surface 146 may be inclined at any angle ϕ necessary to satisfy a specific chain drive geometry and chain pitch elongation. As shown in Figure 9, the inclined root surface angle ϕ is measured from a line 152 passing through the center of roller 128 and the sprocket center to a second line 154 which also passes through the center of roller 128 and Point C. The inclined root surface 146 is normal to the line 154, and the inclined root surface extends radially inward to line 150 where it is tangent to R_i'. In the embodiment being described, the inclined root surface angle ϕ is preferably in the range of about 20° to about 35°.

Figure 12 is an enlarged view of Figure 8 showing the first roller 128 at full engagement in two-point/line contact across the thickness or width of the sprocket tooth profile, and the second roller 130 as the next roller about to mesh with sprocket 112. As with the ISO compliant drive system 10, the chain guide 126 controls and guides a central portion the taut strand 122 except for five unsupported link pitches extending between the chain guide 126 and the engaging roller 128 (and except for the unsupported link pitches extending between the driven sprocket and the chain guide). The taut strand 122 is horizontal when roller 128 is at the 12 o'clock position.

Figure 13 shows the drive sprocket 112 rotated in a clockwise direction (A/2) + ω, as determined by the instant of sprocket engagement by roller 130. A straight line is assumed for the chain span from roller 128 to a chain pin center 156, about which the unsupported span from pin center 156 to engaging roller

130 is considered to rotate. It should be appreciated that the straight line assumption is valid only in a quasi-static model. The amount of movement (or deviation from the straight line assumption) previously mentioned will be a function of the drive dynamics as well as the drive and sprocket geometry.

The sprocket contact at the onset of mesh for roller 130 occurs earlier than for the ISO counterpart, thereby reducing the amount of chordal rise and, just as importantly, allows the initial contact to beneficially occur at a desired pressure angle γ on the engaging flank at Point A. Furthermore, the radial sprocket contact for roller 130, with its contribution to the overall noise level, does not occur until the sprocket rotation places roller 130 at the 12 o'clock position. This is referred to as staged engagement.

Figure 14, an enlarged view of Figure 13, more clearly shows the onset of meshing for roller 130. Just prior to the onset of mesh, roller 128 is assumed to carry the entire taut strand load $F_{TB} + F_{\emptyset}$, which load is shown as force vector arrows. Actually, the arrows represent reaction forces to the taut strand chain force. At the instant of mesh for roller 130, a tangential impact occurs as shown by impact force vector F_{IA}. The tangential impact is not the same as the taut strand chain loading. In particular, impact loading or impact force is related to the impact velocity V_A. It is known that impact occurs during a collision between two bodies, resulting in relatively large forces over a comparatively short interval of time. A radial impact force vector F_{IC} is shown only as an outline in that the radial impact does not occur until the sprocket has rotated sufficiently to place roller 130 at a 12 o'clock position.

Figure 14a shows the same roller positions (solid) for rollers 128 and 130 as shown in Figure 14, but in addition, shows the roller positions (in phantom)

relative to the sprocket profile once roller 130 reaches its two-point/line mesh at the 12 o'clock position. As a result of the pitch mismatch between the chain and sprocket, roller 128 must move to a new position. In particular, as roller 130 moves from initial contact to full mesh, roller 128 progresses forward in its tooth space. Small clearances in the chain joints, however, reduce the amount of forward progression required for roller 128. Also occurring at the onset of meshing is the beginning of the taut strand load transfer from roller 128 to roller 130.

The asymmetrical profile provides for the previously described "staged" meshing. In particular, referring again to Figure 14, the Point A tangential contact occurs at the onset of mesh, with its related impact force F_{IA}. The roller 130 is believed to stay in hard contact with the engaging flank 134 as the sprocket rotation moves the roller into full mesh with its resulting radial contact at Point C. The radial impact force F_{IC} (force vector shown as an outline) does not occur until the sprocket has rotated sufficiently to bring roller 130 into radial contact at Point C.

Figure 14b is an enlarged view of Figure 14, except that sprocket 112 has been rotated to advance roller 130 to the instant of full mesh at the 12 o'clock position. At this instant of full mesh, the radial impact force F_{IC} occurs and the taut strand load transfer is considered to be complete. At the instant of the radial collision by roller 130 at Point C, with its resultant radial impact force F_{IC}, the tangential impact force of F_{IA} has already occurred and is no longer a factor. The time delay ("staged" engagement) between the tangential and radial roller-sprocket collisions effectively spreads the impact energy that must be absorbed during the meshing process over a greater time interval, thereby reducing its contribution to the generated noise level at mesh frequency. Additionally,

it is believed that the present asymmetrical sprocket tooth profile beneficially permits a more gradual taut strand load transfer from a fully engaged roller 128 to a meshing roller 130 as the meshing roller 130 moves from its Point A initial mesh to its full two-point mesh position.

Referring again to Figure 14, the chordal rise (and fall) with the present asymmetrical profile is the perpendicular displacement of the center of roller 130 from the taut strand 122 path as it moves from its initial meshing contact Point A to the mesh position presently occupied by roller 128. It is believed that roller 130 will stay in hard contact with the engaging flank 134 as the roller moves from initial tangential contact to full mesh, and accordingly, the chordal rise is reduced as the distance between Points A and B is increased. As shown in Figure 14, chain pitch P_c is beneficially greater than sprocket 112 chordal pitch P_s.

Referring now to Figure 15, the length of the inclined root surface 146 (Fig. 10) may be reduced to zero (0), thereby eliminating the inclined roof surface 146 and permitting root radius R_i' to be tangent to the root surface and the roller 142 at Point C. That is, R_i' is tangent to a short flat at Point C, and the flat is tangent to R_i'. If the inclined root surface 146 is eliminated, the engaging flank pressure angle γ would generally be in the range of some positive value to zero, but normally not less than zero. The reason is that a negative γ requires chordal pitch reduction so that the roller can exit the sprocket wrap 158 without interfering with R_f.

Figure 16 shows the roller contact to the sprocket 112 profile for all the rollers in the wrap 158. Roller 128 is in full two-point mesh as shown. Line 160 shows the contact point for each of the rollers, as well as the contact progression as the rollers travel around the wrap. The inherent pitch

mismatch between the sprocket and roller chain causes the rollers to climb up the coast side flank as the rollers progress around the sprocket wrap. With the addition of appreciable chordal pitch reduction, the extent to which the rollers climb up the coast side flank in increased.

It is important to note that chordal pitch reduction is required when the pressure angle γ has a negative value. Otherwise, as shown in Figures 16 and 17, roller 162 would interfere with the engaging flank (with a maximum material sprocket and a theoretical pitch [shortest] chain) as it exits the wrap 158 back into the span. Also, the reduced chordal pitch assists the staged mesh as previously mentioned. Figure 17, showing the roller contact progression in the wrap 158, serves also to show why the shallow β' angle and tooth space clearance TSC helps maintain "hard" roller-sprocket contact for the rollers in the wrap.

In addition, the disengaging flank roller seating angle β' (Fig. 9) may be adjusted to have a maximum value which is equal to $\alpha_{min}/2$ or even less. This reduced seating angle β' promotes faster separation when the roller leaves the sprocket and enters the span. This reduced angle β' also allows for the roller in a worn chain to ride up the coast flank surface to a less severe angle as the roller moves around the sprocket in the wrap. Accordingly, chordal pitch reduction, if used in this embodiment, should be a small value.

It is contemplated that the above-described asymmetrical tooth profile features can be altered without substantially deviating from the chain and sprocket meshing kinematics that produce the noise reduction advantages of the present invention. For example, the engaging asymmetrical flank profile could be approximated by an involute form, and the disengaging asymmetrical flank profile could be approximated by a different involute form. Slight changes to the

asymmetrical tooth profiles may be for manufacturing and/or quality control reasons, or simply to improve part dimensioning. These changes are within the scope of the invention as disclosed herein.

In a further embodiment, the engaging flank inclined root surface **146** (Fig. 9) may be replaced with a coast flank inclined root surface **164** as shown in Figure 18. The coast flank inclined root surface **164** provides tooth space clearance (TSC) in the same manner as described above with regard to the inclined root surface **146**. In addition, the engaging flank inclined root surface **164** beneficially moves the roller to a preferred radially outward position as the chain wears.

Alternatively, the coast flank inclined root surface **164** may be included with the engaging flank inclined root surface **146** as shown in Figure 19. The engaging flank and coast flank inclined root surfaces **146, 164** cooperate to provide tooth space clearance (TSC) in the same manner as previously described.

Referring now to Figure 20, any one of the above-described asymmetrical tooth profile embodiments of Figures 9, 15, 18, and 19 may be incorporated into a random-engagement roller chain sprocket **300**. The sprocket **300** is shown as an 18-tooth sprocket. However, the sprocket **300** may have more or less teeth, as desired. The sprocket **300** includes a first group of arbitrarily positioned sprocket teeth **302** each having a profile which incorporates the flank flat **144** shown in Figures 9, 15, 18 and 19. In order to facilitate describing the random engagement roller chain sprocket, reference will be made to the sprocket teeth **302** having the tooth profiles shown in Figures 9 and 15. However, the sprocket teeth **302** may equally incorporate the tooth profiles shown in Figures 18 and 19. The remaining sprocket teeth **304** (sprocket teeth 1, 3, 4, 9, 13, 14, and 16) are randomly positioned around the sprocket and incorporate a tooth profile different from that of the

first group of sprocket teeth 302. As described further below, the first and second groups of sprocket teeth 302, 304 cooperate to reduce chain drive system noise levels below a noise level which either tooth profile used alone would produce.

Figure 21 illustrates an exemplary tooth profile for one of the sprocket teeth 304. An engaging flank 306 and a coast or disengaging flank 308 of an adjacent tooth cooperate to define a tooth space 310 which receives an engaging roller 314 (shown in phantom). The engaging roller 314 has a roller diameter D_1, and is shown fully seated in two-point contact within the tooth space 310. The engaging roller 314 initially contacts the engaging flank 306 at point A' before fully seating in the tooth space at points B and C. Contact points B and C are actually lines that extend axially along each sprocket tooth surface (i.e., in a direction orthogonal to the plane of the drawings).

The engaging flank 306 has a radius R_f which is tangent to a flat surface (not shown) at contact point B. The flat surface, which functions only to facilitate the two-point roller contact (described further below), is normal to the roller 314 at point B. The flat extends radially inward from point B and is tangent to a first root radius R_i. The first root radius R_i may be tangent to a radially outer end of an inclined root surface 316. A maximum root radius R_i must be equal to, or less than, a minimum roller radius to facilitate the fully engaged two-point/line contact at points B and C. Accordingly, a small clearance is defined between the root radius R_i and the roller 314 when the roller 314 is fully seated at points B and C. The flat surface and the inclined root surface 316 necessarily extend inside contact points B and C, respectively, to facilitate the two-point/line roller contact at engagement. A second root radius R_i' is tangent to a radially inner end of the inclined root surface 316 at line 318. The coast flank

has a radius R_f' at a point defined by the roller seating angle β'.

The inclined root surface 316 is a flat surface having a finite length which defines a tooth space clearance (TSC). The tooth space clearance compensates for chain pitch elongation or chain wear by accommodating a specified degree of chain pitch elongation. In other words, the tooth space clearance TSC enables rollers of a worn chain to be maintained in hard contact with the inclined root surface of the sprocket teeth. In addition, the inclined root surface 316 facilitates reducing the radial reaction force thereby reducing the roller radial impact noise contribution to the overall noise level.

The inclined root surface 316 may be inclined at any angle ϕ necessary to satisfy a specific chain drive geometry and chain pitch elongation. The inclined root surface angle ϕ is measured from a line 320 passing through the center of roller 314 and the sprocket center to a second line 322 which also passes through the center of roller 314 and through contact point C. The inclined root surface 316 is normal to the line 322, and the inclined root surface extends radially inward to line 318 where it is tangent to R_i'. In the embodiment being described, the inclined root surface angle ϕ is preferably in the range of about 20° to about 35°. In sum, the tooth profile for the sprocket teeth 304 may be substantially the same as the tooth profile shown in Figures 9 and 15 except that the tooth profile 304 does not have an engaging flank flat extending above (i.e. radially outward of) contact point A'.

As with the tooth profile of Figures 9 and 15, the engaging flank inclined root surface 316 may be replaced with a coast flank inclined root surface as in Figure 18. That is, the tooth profile 304 may be substantially identical to the sprocket 112 shown in Figure 18 from contact point C to the outer diameter of

the coast flank 308. The coast flank inclined root surface provides tooth space clearance (TSC) in the same manner as the inclined root surface 316. In addition, the coast flank inclined root surface beneficially moves the roller to a preferred radially outward position as the chain wears. Alternatively, the coast flank inclined root surface may be included with the engaging flank inclined root surface 316 as in Figure 19. Thus, the tooth profile 304 may be substantially identical to the sprocket 112 shown in Figure 19 from contact point C to the outer diameter of the coast flank 138.

Pitch mismatch is inherent in a chain/sprocket interface except at one condition - the theoretical condition which is defined as a chain at its shortest pitch (shortest being theoretical pitch) and a maximum material sprocket. This theoretical condition therefore defines one limit (zero, or no pitch mismatch) of the tolerance range of the pitch mismatch relationship of chain and sprocket. The other limit is defined when a longest "as built" chain is used with a sprocket at minimum material conditions - or in other words, a sprocket having a minimum profile. This limit produces the greatest amount of pitch mismatch. The pitch mismatch range is therefore determined by the part feature tolerances.

Additional pitch mismatch may be introduced to facilitate a greater time delay between tangential contact at point A (for tooth profile 302) and tangential contact at point A' (for tooth profile 304). That is, varying the time at which roller-to-sprocket contact occurs for each tooth profile 302, 304 results in reduced mesh frequency noise because the point and rhythm of the initial roller-to-sprocket contact is altered. The time delay between the roller-to-sprocket contact at points A and A' may be increased by increasing the mismatch between the chain pitch and sprocket pitch.

Additional pitch mismatch may also be introduced to facilitate a "staged" roller contact for each tooth profile 302, 304. That is, additional pitch mismatch increases the time delay between initial tangential contact and the fully seated radial contact for each tooth profile 302, 304. It should be appreciated that staged contact is greater for the tooth profile 302 than for the tooth profile 304 due to the flank flat 144 which causes initial contact to occur higher up on the engaging flank of the sprocket teeth 302.

The sprocket chordal pitch is necessarily shorter than the chain pitch to facilitate the "staged" roller-tooth contact. In addition, chordal pitch reduction also provides roller-to-flank clearance as the roller exits the sprocket wrap back into the strand. Added chordal pitch reduction, when used, is preferably in the range of 0.005-0.030 mm.

The staged roller contact for each tooth profile 302, 304 may be further assisted by providing sprocket tooth pressure angles γ that are substantially less than the ISO standard. Pressure angles γ equal to or very close to zero (0), or even negative pressure angles, are contemplated. For instance, Figure 22 illustrates one embodiment of a random engagement sprocket 330 wherein the tooth profiles 302, 304 have the same, or at least substantially the same, pressure angles γ (thus, the profiles 302, 304 have the same or at least substantially the same roller seating angles β_{302} and β_{304}). In Figure 22, the pressure angle for both profiles 302, 304 is zero or a positive value. Thus, γ_{min} for both profiles 302, 304 may be 0°, and γ_{max} for both profiles 302, 304 is equal to some value less that the ISO minimum pressure angle γ. As a result, initial roller-to-sprocket contact occurs at point A followed by subsequent radial contact at points B and C for the tooth profile 302 of sprocket 330. And, initial roller-

to-sprocket contact occurs at point A' followed by subsequent radial contact at points B and C for the tooth profile 304 of sprocket 330. The sprocket 330 may, or may not incorporate additional chordal pitch reduction, and may, or may not incorporate tooth space clearance (TSC), as described above.

With reference to Figure 23, a second embodiment of a random engagement sprocket 340 is shown wherein the pressure angles γ for the tooth profiles 302, 304 are different. That is, both pressure angles γ are either a positive value or zero, but the pressure angle γ for tooth profile 302 is smaller than the pressure angle γ for tooth profile 304 (and the roller seating angle β_{302} is greater than the roller seating angle β_{304}). Thus, γ_{min} may be equal to 0° for the tooth profile 302, γ_{min} may be equal to +4 ° for the tooth profile 304, and γ_{max} for both profiles 302, 304 will always be less than the ISO minimum pressure angle (the feature tolerance band or range for γ_{min} and γ_{max} is the same for both tooth profiles 302, 304). As a result, initial roller-to-sprocket contact occurs at point A followed by subsequent radial contact at points B and C for the tooth profile 302 of sprocket 340. And, initial roller-to-sprocket contact occurs at point A' followed by subsequent radial contact at points B' and C for the tooth profile 304 of sprocket 340. The sprocket 340 may or may not incorporate additional chordal pitch reduction, and may or may not incorporate tooth space clearance (TSC), as desired.

Alternatively, the pressure angle γ_{min} for the profile 302 may always be a negative value, while the pressure angle γ_{min} for profile 304 may always be a positive value or zero. For instance, γ_{min} may be equal to -3° for the tooth profile 302, γ_{min} may be equal to +3° for the tooth profile 304, and γ_{max} for both profiles 302, 304 will always be less than the ISO minimum pressure angle. With this embodiment, additional chordal pitch

reduction will always be included, however, tooth space clearance may or may not be included.

Figure 24 shows the engagement path of a roller 342 fully seated in the sprocket tooth 302, and an engagement path of the roller 314 engaging the adjacent sprocket tooth 304 of the sprocket 340. At the instant of initial contact for roller 314, the chain load transfer from roller 342 to roller 314 begins, until at full engagement, roller 314 will substantially carry the full chain load. Also, as roller 314 moves from initial contact to full mesh, roller 342 will progress forward in its tooth space. Small clearances in the chain joints, however, will reduce the amount of forward progression required for Roller 342. Since impact energy is reduced with this profile geometry, the load transfer is more gradual than with the standard ISO profile.

The references ℓ_A and ℓ_B in Figure 24 serve to show the amount of "staging" for each profile from the initial contact to the two-point contact at full mesh. The reference ℓ_A indicates the linear distance separating the initial contact point A from the contact point B for the profile 302, and the reference ℓ_B indicates the linear distance separating the initial contact point A' from the contact point B' for the profile 304. The value ℓ_A - ℓ_B is a measure of the initial (tangential) contact time delay from profile 304 to profile 302, which is illustrated further in Figures 25 and 26. The time delay beneficially increases with chain wear for the embodiment incorporating different pressure angles (Figure 23) in that the tooth profile 304 "falls away" faster than does the tooth profile 302. The delay increases with increasing chain pitch or pitch mismatch.

Figures 25 and 26 illustrate the meshing delay between the tooth profiles 302, 304. In particular, as shown in Figure 25, the sprocket 340 has a roller 344

fully-seated in two-point contact with a sprocket tooth incorporating the tooth profile 302. The roller 342 is shown at the instant of initial tangential contact at point A of a second sprocket tooth also incorporating the tooth profile 302. The roller 314 is the next roller in the span and will mesh with a sprocket tooth incorporating the tooth profile 304. The sprocket 340 must rotate through an angle τ for roller 342 to move from its initial contact position at point A to full mesh, seated in two-point contact with the tooth profile 302 at a 12 o'clock position.

With reference now to Figure 26, the sprocket 340 is shown rotated in a clockwise direction until roller 314 is at the instant of initial tangential contact at point A' of the tooth profile 304. The sprocket 340 must now rotate through a smaller angle κ for roller 314 to fully seat in two-point contact with the tooth profile 304 at a 12 o'clock position. That is, the staged contact for the tooth profile 302 is greater than for the tooth profile 304 due to the flank flat 144 as well as the difference in engaging side roller seating angles β_{302} and β_{304} which cause initial contact to occur higher up on the engaging flank of the sprocket teeth 302. Thus, the sprocket 340 must rotate through an additional angle $\tau - \kappa$ in order for the roller 314 to fully seat.

As shown in Figure 20, the two sets of tooth profiles 302, 304 are arranged in a random pattern in order to modify the meshing impact frequency by altering the point and rhythm of initial roller-to-sprocket contact. However, the two sets of tooth profiles 302, 304 could be arranged in many different random patterns. Further, it is also contemplated that the two sets of tooth profiles 302, 304 could be arranged in many regular patterns that would work equally as well. In all cases, the arrangement of two sets of different tooth profiles on a sprocket provides a means for

breaking up the mesh frequency impact noise normally associated with and induced by a full complement of substantially identically shaped sprocket teeth. The mesh frequency noise reduction is achieved by altering the point and rhythm of initial roller-to-sprocket contact.

The crankshaft sprocket, generally the smallest sprocket in the chain drive, is usually the major noise contributor. The typically larger driven camshaft sprocket, however, will also contribute to the generated noise levels, but generally to a lesser extent than the crankshaft sprocket. However, the driven sprocket, particularly if it is nearly the same size or smaller than the driving sprocket, may be the prime noise generator, as in the case with balance shaft sprockets and pump sprockets. Thus, the features of the present invention may also be used advantageously with camshaft or driven sprockets as well.

It should be appreciated that the tooth profile features of Figures 20-26 can be altered slightly without substantially deviating from the chain and sprocket meshing kinematics that produce the noise reduction advantages of the present invention. For example, the engaging asymmetrical flank profile could be approximated by an involute form, and the disengaging asymmetrical flank profile could be approximated by a different involute form. Slight changes to the profile may be done for manufacturing and/or quality control reasons - or simply to improve part dimensioning.

The invention has been described with reference to the preferred embodiments. Obviously, modifications will occur to others upon a reading and understanding of this specification and this invention is intended to include same insofar as they come within the scope of the appended claims or the equivalents thereof.

ROLLER CHAIN SPROCKETS

ROLLER SEATING & PRESSURE ANGLES

ISO

NO. OF TEETH Z	TOOTH ANGLE A	ROLLER SEATING ANGLE *ALPHA*		PRESSURE ANGLE *GAMMA*	
		MINIMUM	MAXIMUM	MAXIMUM	MINIMUM
18	20.000	115.00	135.00	22.500	12.500
19	18.947	115.26	135.26	22.895	12.895
20	18.000	115.50	135.50	23.250	13.250
21	17.143	115.71	135.71	23.571	13.571
22	16.364	115.91	135.91	23.864	13.864
23	15.652	116.09	136.09	24.130	14.130
24	15.000	116.25	136.25	24.375	14.375
25	14.400	116.40	136.40	24.600	14.600
26	13.846	116.54	136.54	24.808	14.808
27	13.333	116.67	136.67	25.000	15.000
28	12.857	116.79	136.79	25.179	15.179
29	12.414	116.90	136.90	25.345	15.345
30	12.000	117.00	137.00	25.500	15.500

TABLE 1
PRIOR ART

ROLLER CHAIN SPROCKETS

ROLLER SEATING & PRESSURE ANGLES

	ISO				ASYMMETRICAL					
					PROFILE 1		PROFILE 2		PROFILE 3	
NO. OF TEETH Z	ROLLER SEATING ANGLE ALPHA (MIN)	ROLLER SEATING ANGLE ALPHA (MAX)	ALPHA (MAX) /2	PRESSURE ANGLE GAMMA (MIN)	BETA (MAX)	PRESSURE ANGLE GAMMA (MIN)	BETA (MAX)	PRESSURE ANGLE GAMMA (MIN)	BETA (MAX)	PRESSURE ANGLE GAMMA (MIN)
18	115.00	135.00	67.50	12.500	73.75	6.25	80.00	0	82.00	-2
19	115.26	135.26	67.63	12.895	74.08	6.45	80.53	0	82.53	-2
20	115.50	135.50	67.75	13.250	74.38	6.63	81.00	0	83.00	-2
21	115.71	135.71	67.86	13.571	74.64	6.79	81.43	0	83.43	-2
22	115.91	135.91	67.95	13.864	74.89	6.93	81.82	0	83.82	-2
23	116.09	136.09	68.04	14.130	75.11	7.07	82.17	0	84.17	-2
24	116.25	136.25	68.13	14.375	75.31	7.19	82.50	0	84.50	-2
25	116.40	136.40	68.20	14.600	75.50	7.30	82.80	0	84.80	-2
26	116.54	136.54	68.27	14.808	75.67	7.40	83.08	0	85.08	-2
27	116.67	136.67	68.33	15.000	75.83	7.50	83.33	0	85.33	-2
28	116.79	136.79	68.39	15.179	75.98	7.59	83.57	0	85.57	-2
29	116.80	136.90	68.45	15.345	76.12	7.67	83.79	0	85.79	-2
30	117.00	137.00	68.50	15.500	76.25	7.75	84.00	0	86.00	-2

TABLE 2

Having thus described the preferred embodiments, the invention is now claimed to be:

1. A sprocket comprising:

a first plurality of sprocket teeth each having a first tooth profile including a first engaging flank with a first contact point at which a roller initially contacts said first engaging flank;

a second plurality of sprocket teeth each having a second tooth profile including a second engaging flank with a second contact point at which a roller initially contacts said second engaging flank; and

said first tooth profile including a flank flat on said first engaging flank which facilitates spacing said first contact point on said first engaging flank relative to said second contact point on said second engaging flank to alter an initial roller-to-first engaging flank contact relative to an initial roller-to-second engaging flank contact.

2. The sprocket of claim 1, wherein said flank flat facilitates spacing said first contact point higher up on said first engaging flank relative to said second contact point on said second engaging flank.

3. The sprocket of claim 1, wherein said flank flat is tangent to an engaging flank radius at a radially outer end thereof, and tangent to a root radius at a radially inner end thereof.

4. The sprocket of claim 1, wherein

said first plurality of sprocket teeth including a first engaging side root radius,

said second plurality of sprocket teeth including a second engaging side root radius, and

an engaging side inclined root surface tangent to at least one of said first engaging side root radius and said second engaging side root radius to provide tooth space clearance.

5. The sprocket of claim 4, wherein

said first plurality of sprocket teeth further including an adjacent first disengaging flank,

said second plurality of sprocket teeth further including an adjacent second disengaging flank, and

a disengaging side inclined root surface associated with at least one of said adjacent first disengaging flank and said adjacent second disengaging flank to also provide tooth space clearance.

6. The sprocket of claim 4, wherein said first engaging side root radius and said second engaging side root radius are both less that a roller radius.

7. The sprocket of claim 1, wherein

said first plurality of sprocket teeth including an adjacent first disengaging flank,

said second plurality of sprocket teeth including an adjacent second disengaging flank, and

a disengaging side inclined root surface associated with at least one of said adjacent first disengaging flank and said adjacent second disengaging flank to provide tooth space clearance.

8. The sprocket of claim 1, further including a roller chain having rollers which mesh with said first plurality of sprocket teeth and said second plurality of sprocket teeth, said roller chain having a chain pitch and the sprocket having a chordal pitch which is less than said chain pitch to facilitate a

staged contact between said rollers and said first and said second engaging flanks.

9. The sprocket of claim 1, wherein an engaging flank pressure angle γ of said first plurality of sprocket teeth is substantially the same as an engaging flank pressure angle γ of said second plurality of sprocket teeth.

10. The sprocket of claim 9, wherein a minimum engaging flank pressure angle γ_{min} of said first and said second plurality of sprocket teeth is greater than or equal to zero, and a maximum engaging flank pressure angle γ_{max} of said first and said second plurality of sprocket teeth is less than an ISO-606 minimum engaging flank pressure angle γ_{ISOmax}.

11. The sprocket of claim 1, wherein

a minimum engaging flank pressure angle γ_{min} of said first plurality of sprocket teeth is greater than or equal to zero,

a minimum engaging flank pressure angle γ_{min} of said second plurality of sprocket teeth is greater than said minimum engaging flank pressure angle of said first plurality of sprocket teeth, and

a maximum engaging flank pressure angle γ_{max} of said first and said second plurality of sprocket teeth is less than an ISO-606 minimum engaging flank pressure angle γ_{ISOmax}.

12. The sprocket of claim 1, wherein

a minimum engaging flank pressure angle γ_{min} of said first plurality of sprocket teeth is a negative value,

a minimum engaging flank pressure angle γ_{min} of said second plurality of sprocket teeth is a positive value, and

a maximum engaging flank pressure angle γ_{max} of said first and said second plurality of sprocket teeth is less than an ISO-606 minimum engaging flank pressure angle γ_{ISOmax}.

13. A unidirectional roller chain drive system comprising:

a driving sprocket having sprocket teeth with engaging flanks and coast flanks, the engaging flanks cooperating with the coast flanks of adjacent teeth to define asymmetrical tooth spaces between the sprocket teeth;

a driven sprocket having sprocket teeth with engaging flanks and coast flanks, the engaging flanks cooperating with the coast flanks of adjacent teeth to define asymmetrical tooth spaces between the sprocket teeth;

a roller chain having rollers in engaging contact with the driving sprocket and the driven sprocket; and

at least one of said driving sprocket and said driven sprocket including:

a first plurality of sprocket teeth each having a first engaging flank with a first contact point at which a roller initially contacts said first engaging flank,

a second plurality of sprocket teeth each having a second engaging flank with a second contact point at which a roller initially contacts said second engaging flank, and

said first engaging flank including a flank flat which facilitates phasing a frequency of initial roller-to-first engaging flank contacts relative to initial roller-to-second engaging flank contacts to alter the rhythm of said initial roller-to-first

engaging flank and said roller-to-second engaging flank contacts.

14. The roller chain drive system of claim 13, wherein said flank flat facilitates spacing said first contact point higher up on said first engaging flank relative to said second contact point on said second engaging flank.

15. The roller chain drive system of claim 13, wherein said flank flat is tangent to an engaging flank radius at a radially outer end thereof, and tangent to a root radius at a radially inner end thereof.

16. The roller chain drive system of claim 13, wherein

said first plurality of sprocket teeth including a first engaging side root radius,

said second plurality of sprocket teeth including a second engaging side root radius, and

an engaging side inclined root surface tangent to at least one of said first engaging side root radius and said second engaging side root radius to provide tooth space clearance.

17. The roller chain drive system of claim 16, wherein

said first plurality of sprocket teeth further including an adjacent first disengaging flank,

said second plurality of sprocket teeth further including an adjacent second disengaging flank, and

a disengaging side inclined root surface associated with at least one of said adjacent first disengaging flank and said adjacent second disengaging flank to also provide tooth space clearance.

18. The roller chain drive system of claim 16, wherein said first engaging side root radius and said second engaging side root radius are both less that a roller radius.

19. The roller chain drive system of claim 13, wherein

said first plurality of sprocket teeth including an adjacent first disengaging flank,

said second plurality of sprocket teeth including an adjacent second disengaging flank, and

a disengaging side inclined root surface associated with at least one of said adjacent first disengaging flank and said adjacent second disengaging flank to provide tooth space clearance.

20. The roller chain drive system of claim 13, further including a roller chain having rollers which mesh with said first plurality of sprocket teeth and said second plurality of sprocket teeth, said roller chain having a chain pitch and the sprocket having a chordal pitch which is less than said chain pitch to facilitate a staged contact between said rollers and said first and said second engaging flanks.

21. The roller chain drive system of claim 13, wherein an engaging flank pressure angle γ of said first plurality of sprocket teeth is substantially the same as an engaging flank pressure angle γ of said second plurality of sprocket teeth.

22. The roller chain drive system of claim 21, wherein a minimum engaging flank pressure angle γ_{min} of said first and said second plurality of sprocket teeth is greater than or equal to zero, and a maximum engaging flank pressure angle γ_{max} of said first and said

second plurality of sprocket teeth is less than an ISO-606 minimum engaging flank pressure angle γ_{ISOmax}.

23. The roller chain drive system of claim 13, wherein

a minimum engaging flank pressure angle γ_{min} of said first plurality of sprocket teeth is greater than or equal to zero,

a minimum engaging flank pressure angle γ_{min} of said second plurality of sprocket teeth is greater than said minimum engaging flank pressure angle of said first plurality of sprocket teeth, and

a maximum engaging flank pressure angle γ_{max} of said first and said second plurality of sprocket teeth is less than an ISO-606 minimum engaging flank pressure angle γ_{ISOmax}.

24. The roller chain drive system of claim 13, wherein

a minimum engaging flank pressure angle γ_{min} of said first plurality of sprocket teeth is a negative value,

a minimum engaging flank pressure angle γ_{min} of said second plurality of sprocket teeth is a positive value, and

a maximum engaging flank pressure angle γ_{max} of said first and said second plurality of sprocket teeth is less than an ISO-606 minimum engaging flank pressure angle γ_{ISOmax}.

25. A sprocket comprising:

a first plurality of sprocket teeth each having a first engaging flank with a first contact point at which a roller initially contacts said first engaging flank;

a second plurality of sprocket teeth each having a second engaging flank with a second contact

point at which a roller initially contacts said second engaging flank; and

said first engaging flank including a flank flat which facilitates phasing a frequency of initial roller-to-first engaging flank contacts relative to initial roller-to-second engaging flank contacts to alter the rhythm of said initial roller-to-first engaging flank and said roller-to-second engaging flank contacts.

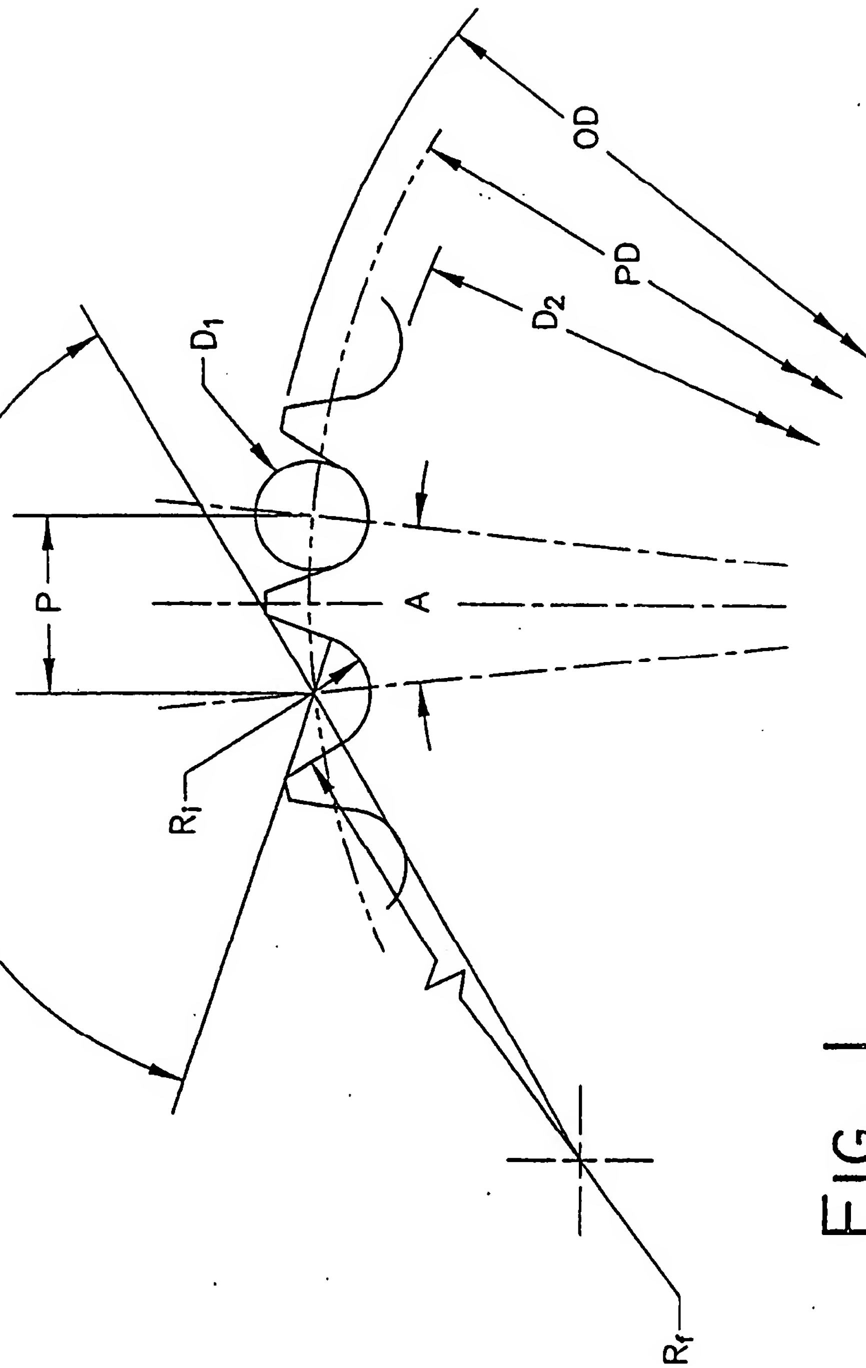


FIG. 1
PRIOR ART

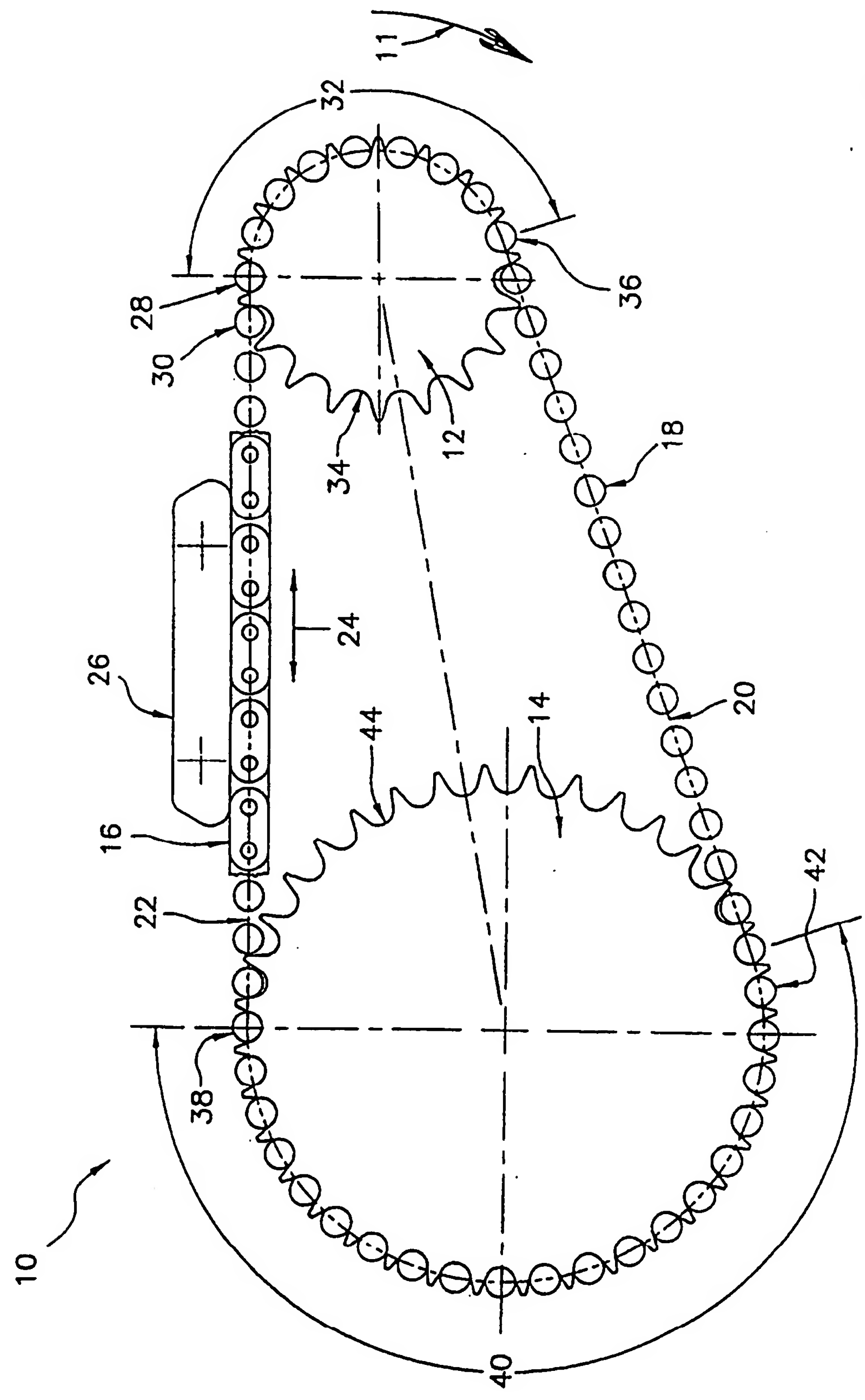


FIG. 2
PRIOR ART

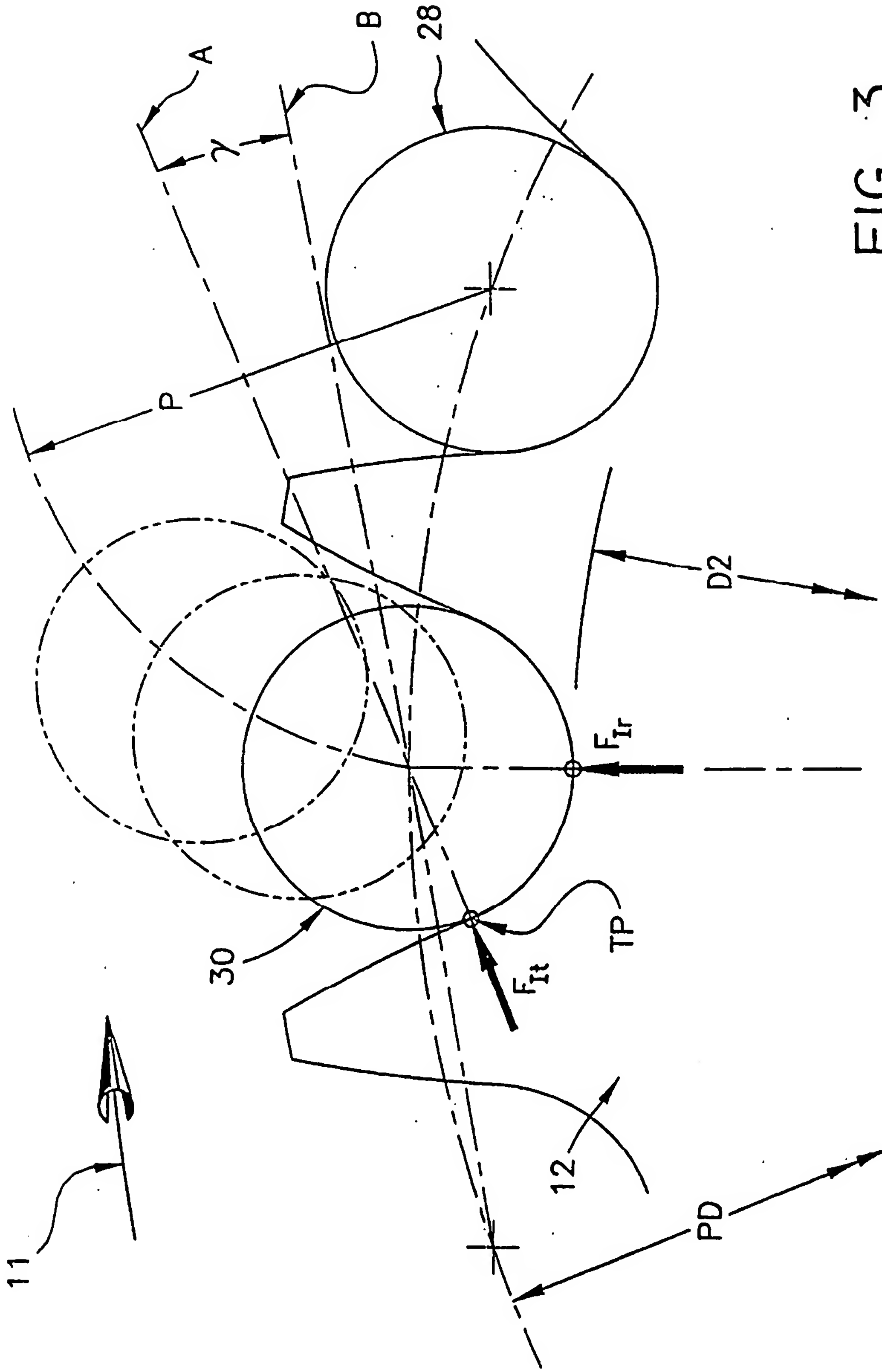


FIG. 3
PRIOR ART

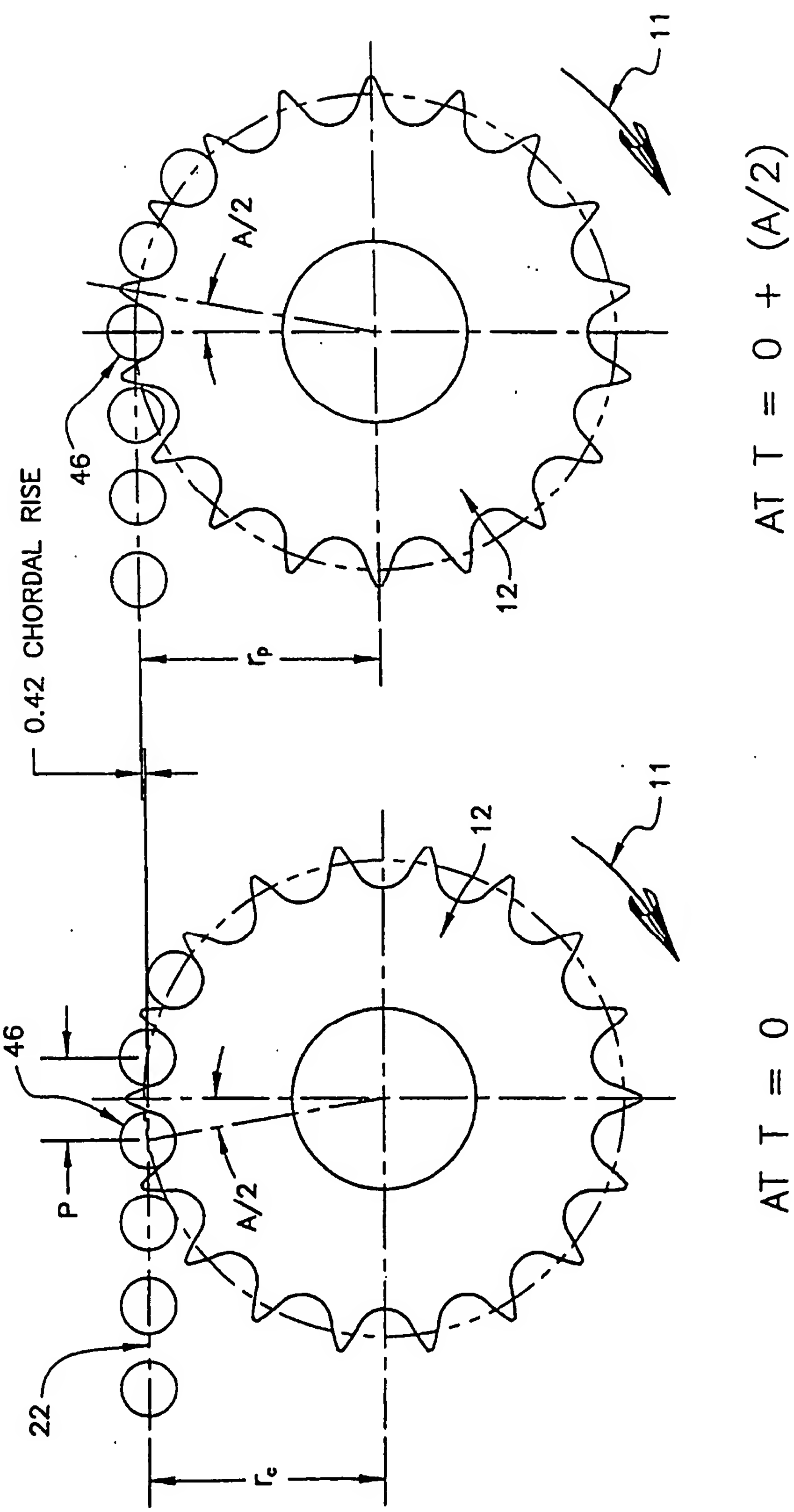


FIG. 4
PRIOR ART

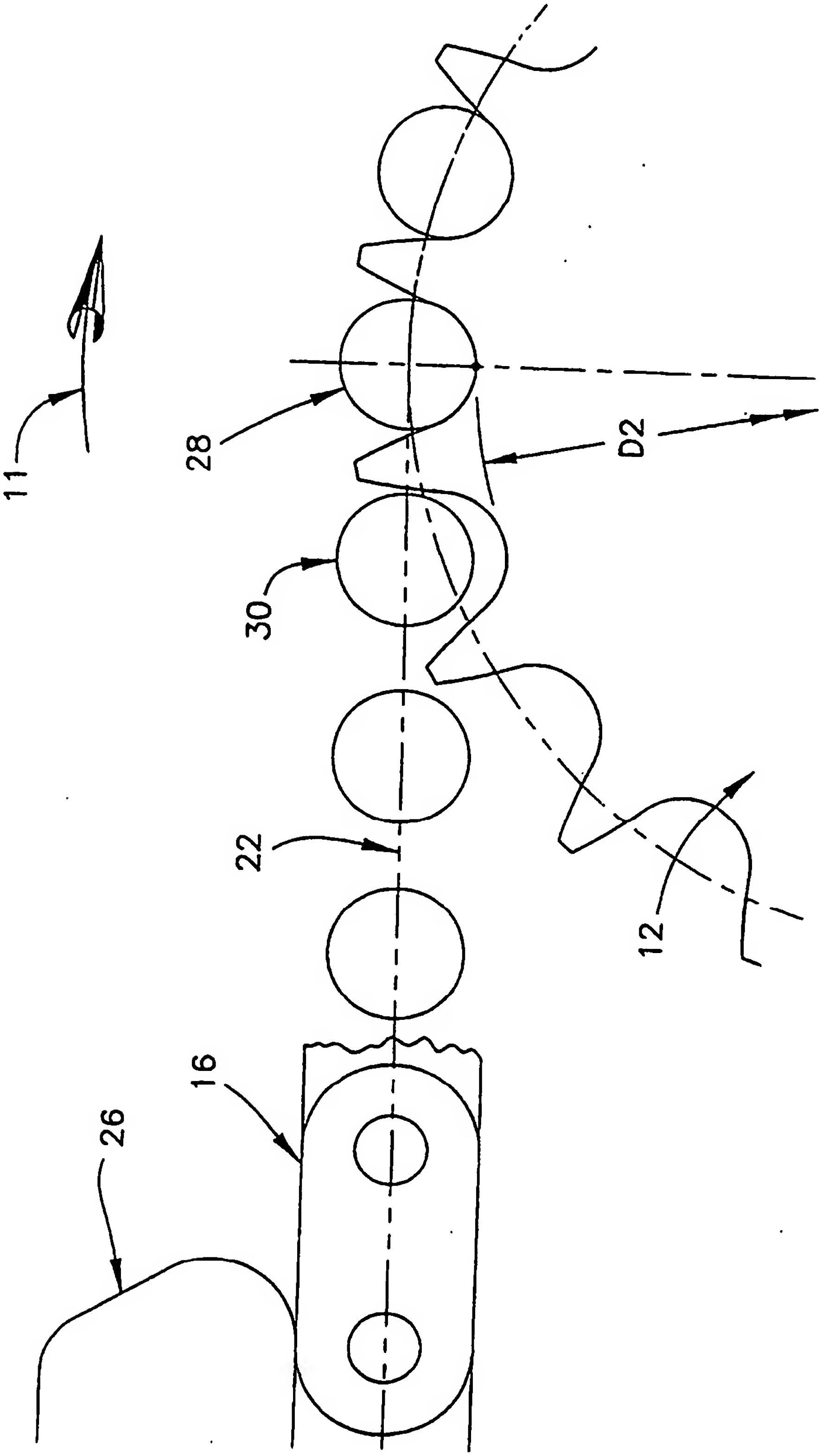


FIG. 5
PRIOR ART

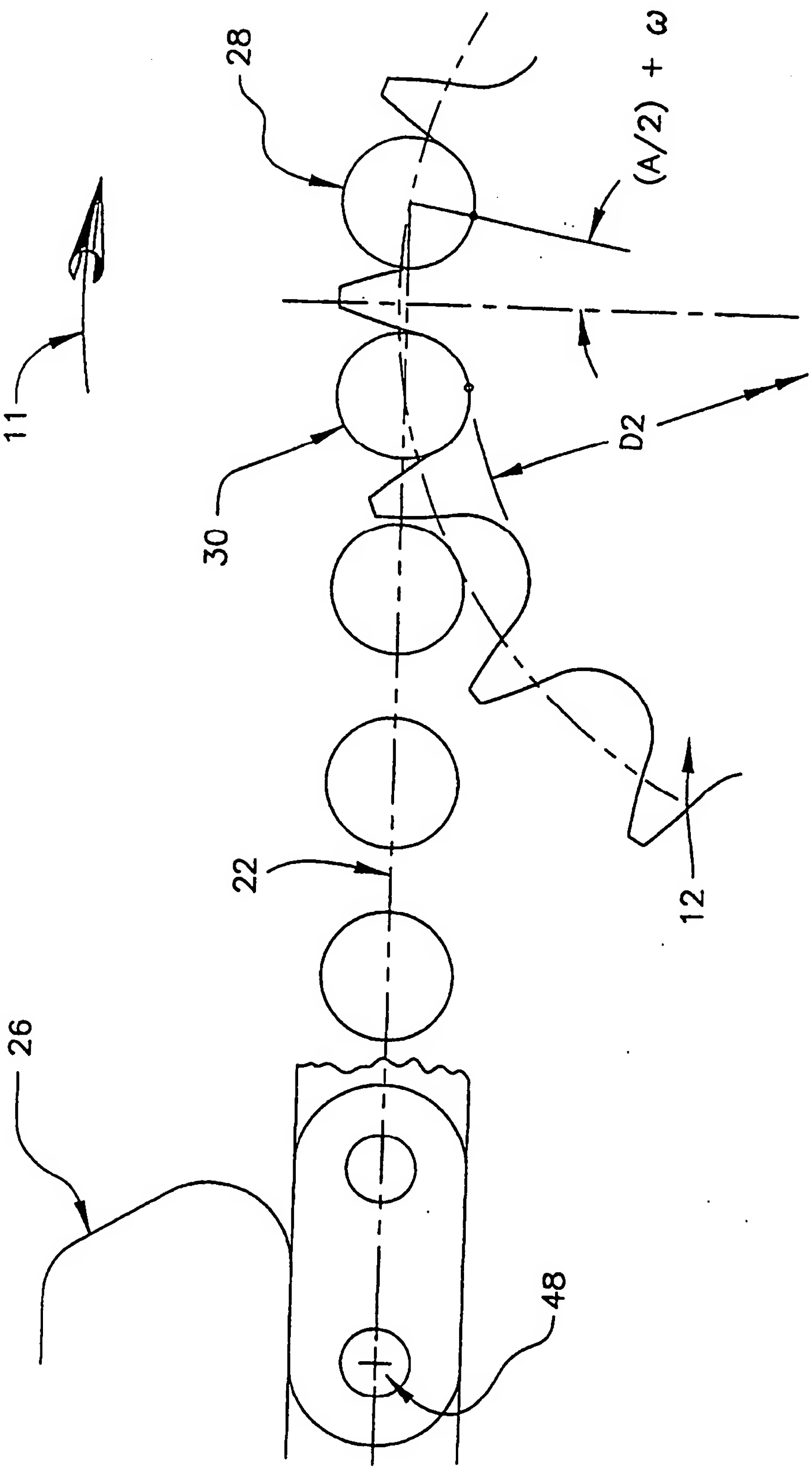


FIG. 6
PRIOR ART

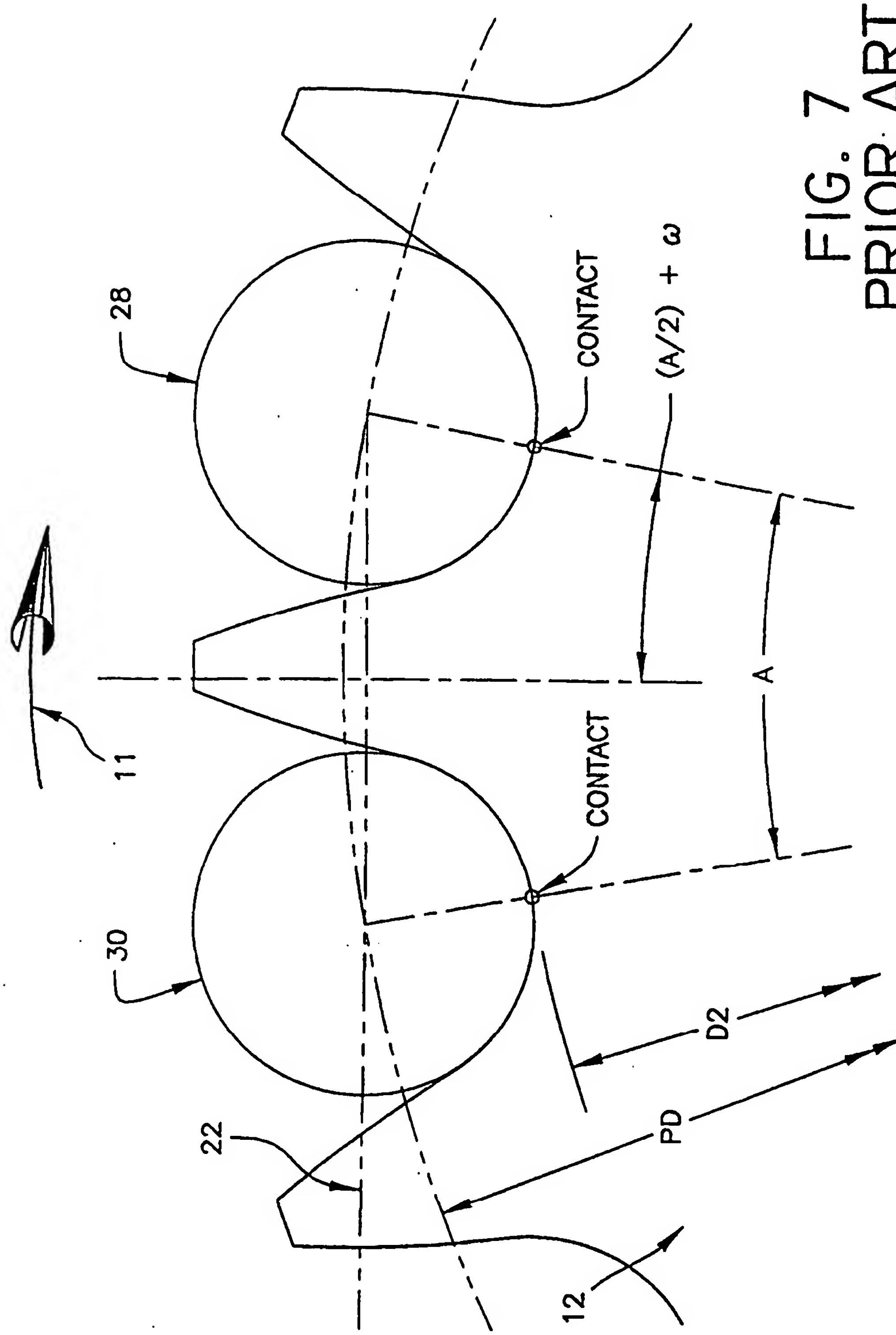


FIG. 7
PRIOR ART

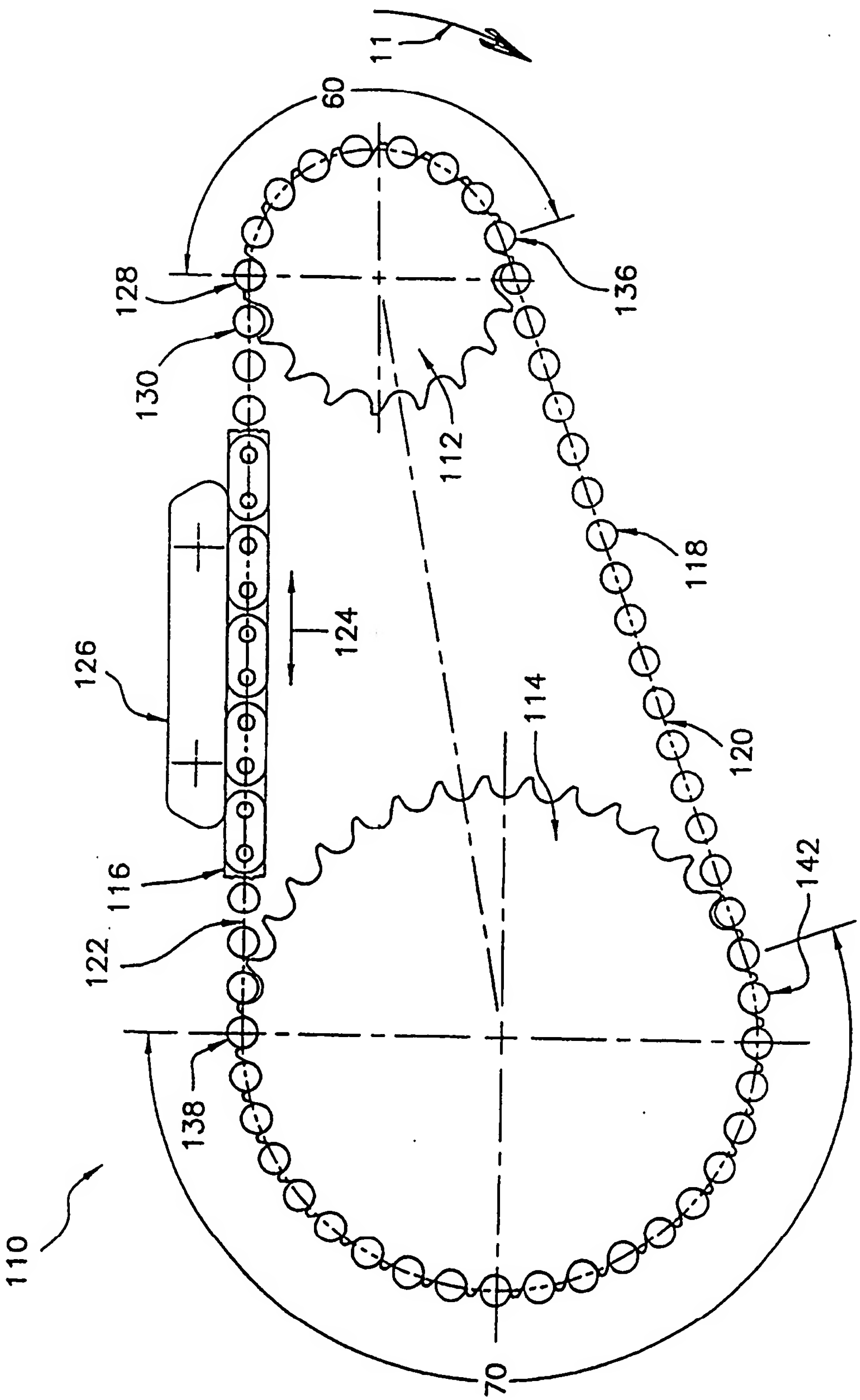
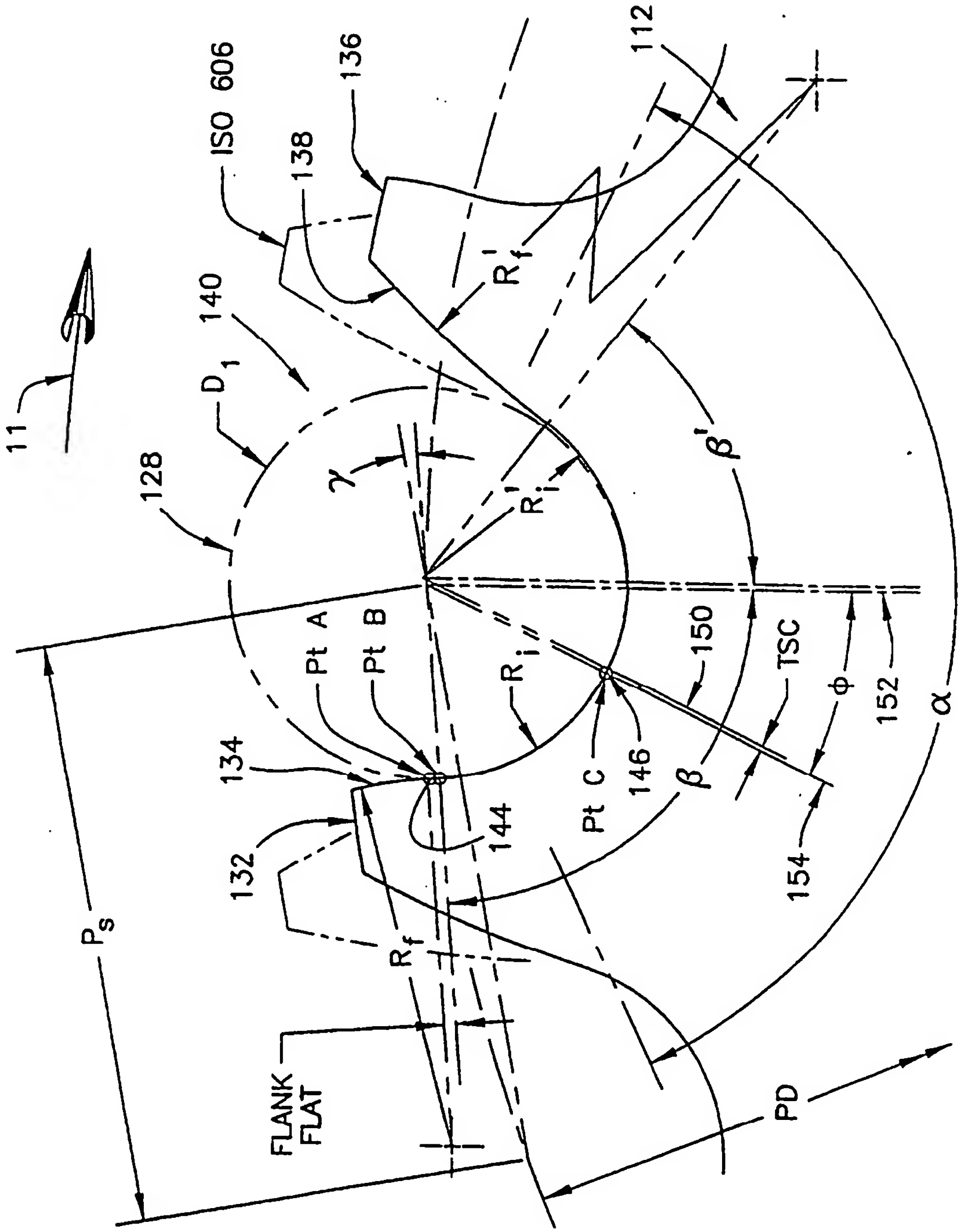


FIG. 8


11
128
D1
140
ISO 606
138
136
112
R'f
R'i
γ
β'
Pt A
Pt B
Ri
150
TSC
φ
152
α
134
132
144
Pt C
146
β
154
Ps
Rf
FLANK FLAT
PD

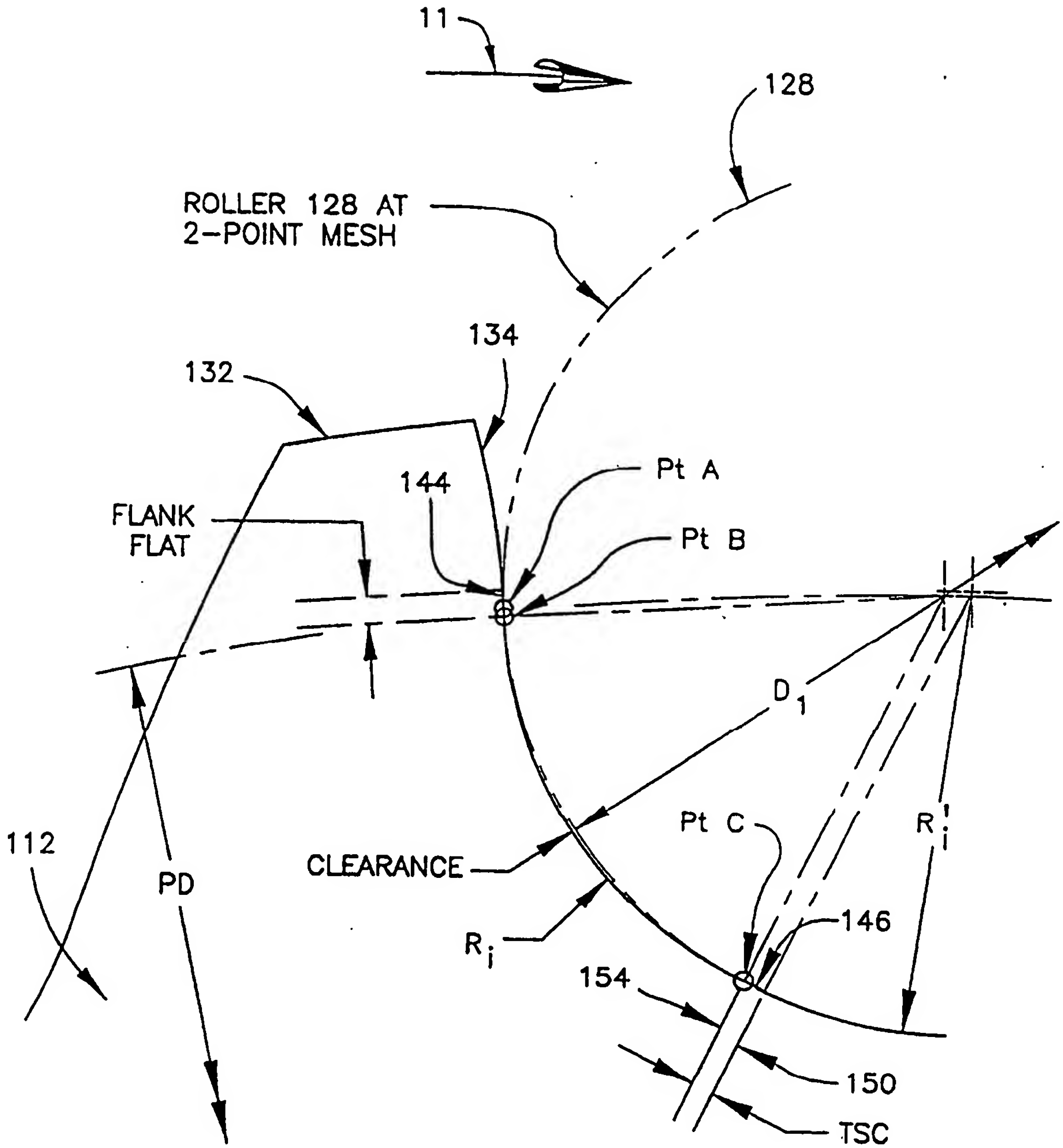

11
128
ROLLER 128 AT
2-POINT MESH
134
132
144
Pt A
FLANK
FLAT
Pt B
D1
112
PD
CLEARANCE
Pt C
R'i
Ri
146
154
150
TSC

FIG. 10

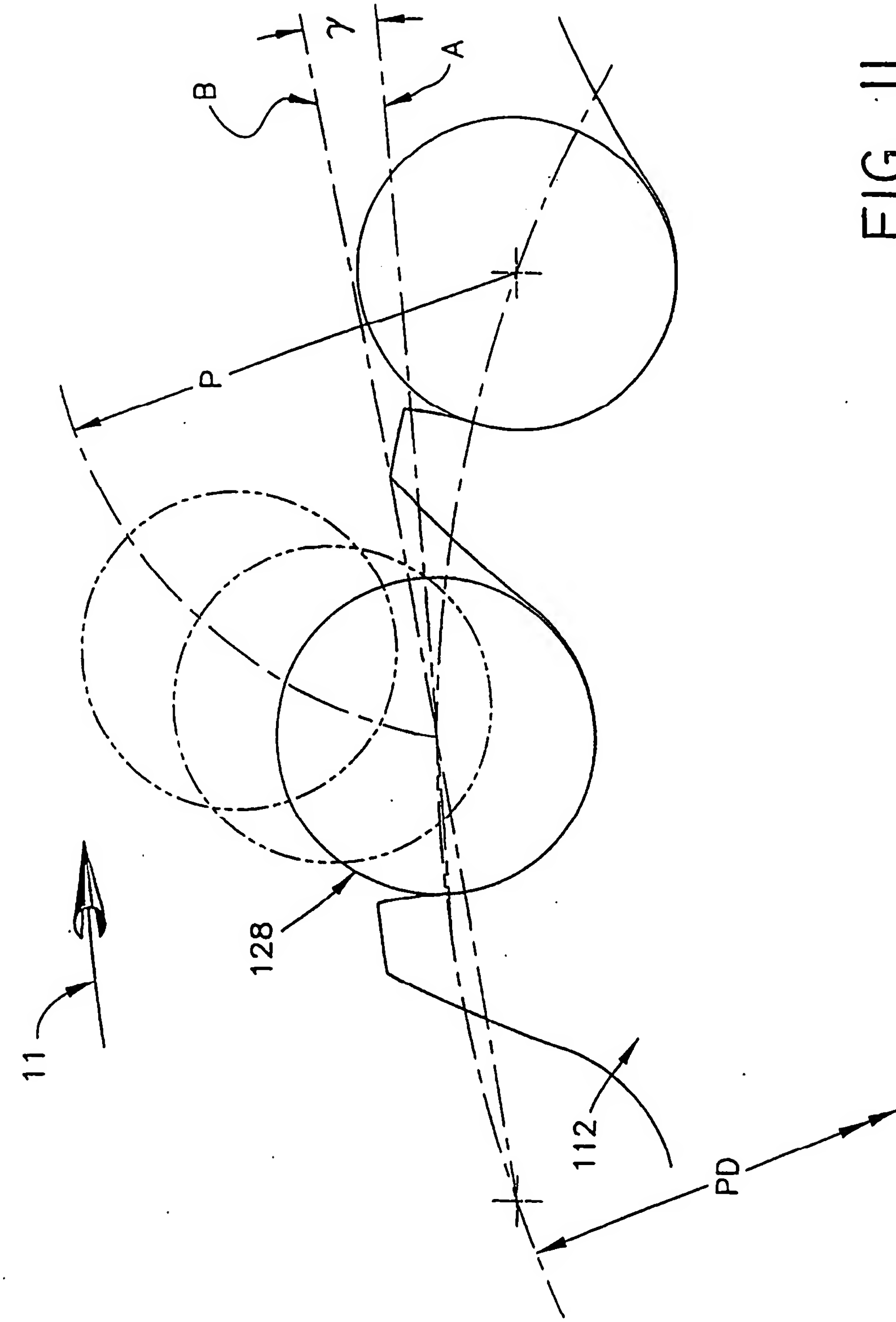


FIG. 11

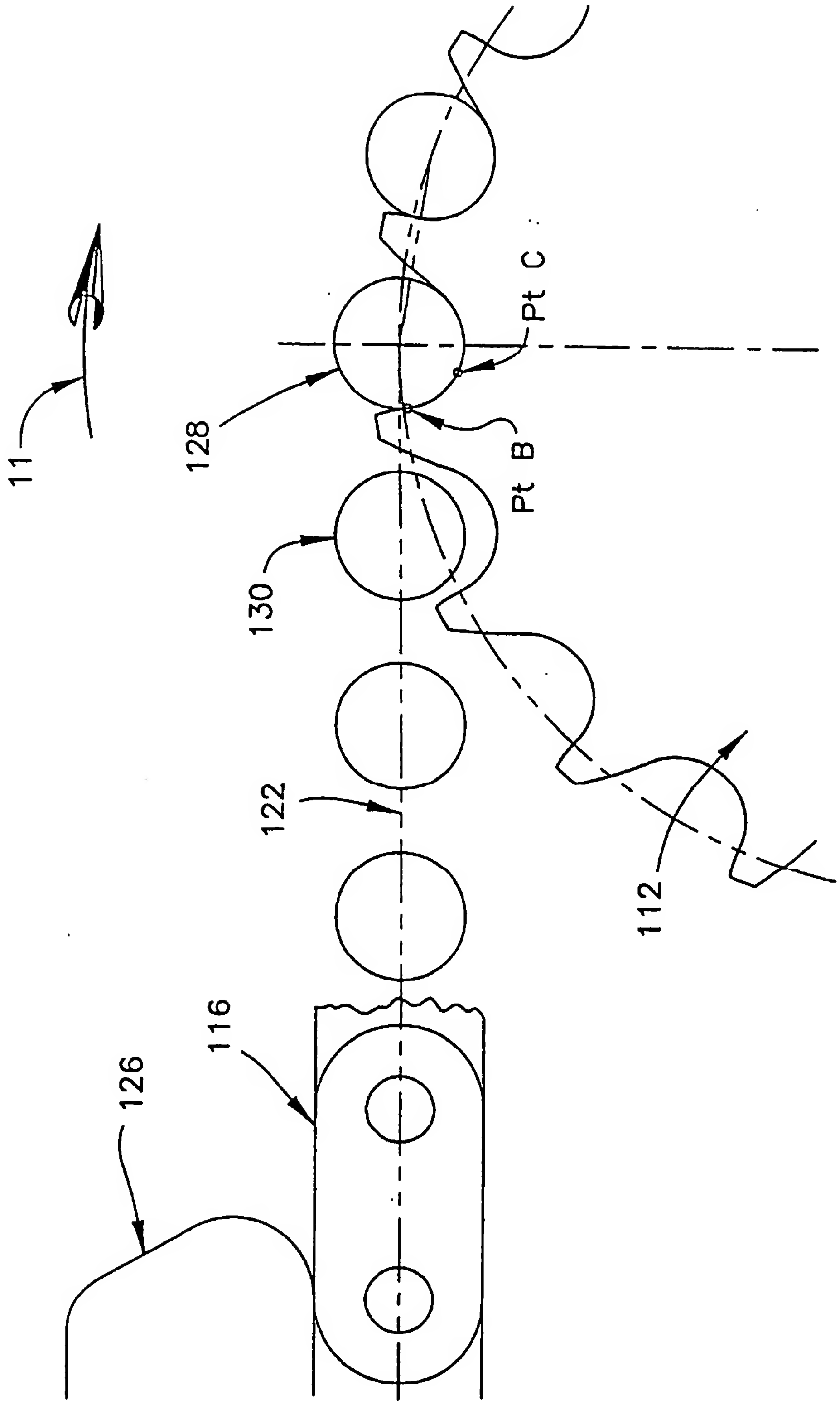


FIG. 12

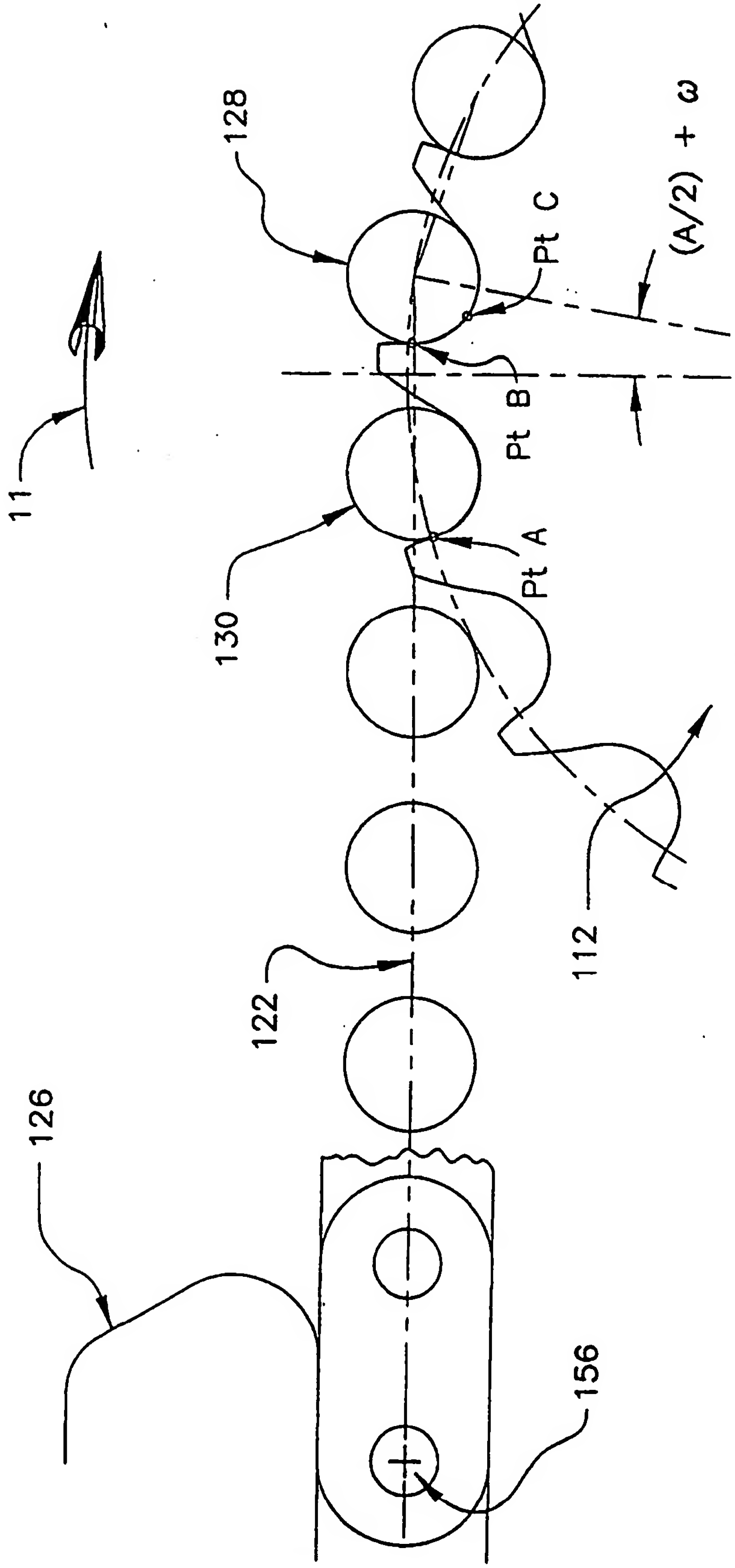


FIG. 13

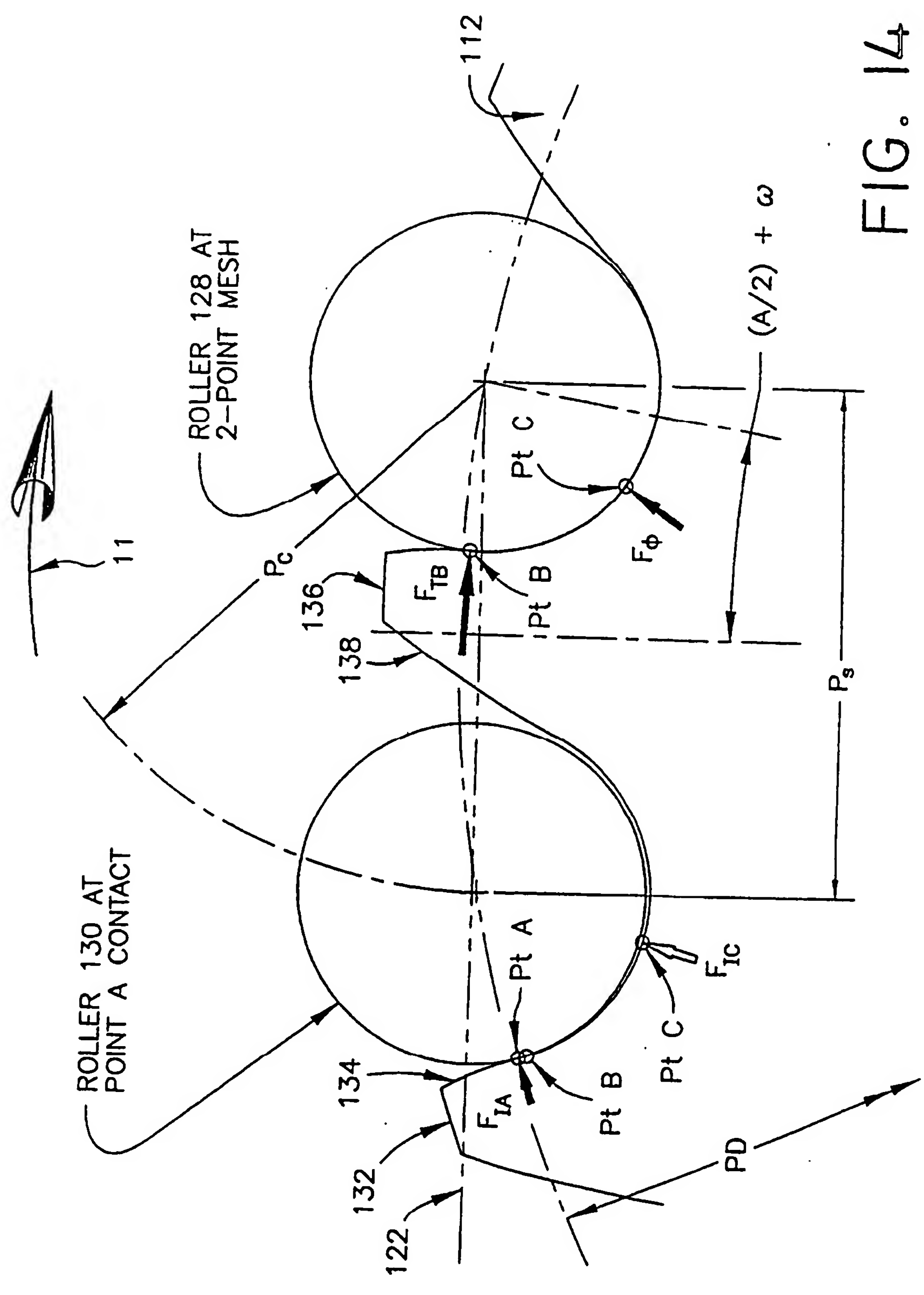

ROLLER 128 AT 2-POINT MESH
112
11
Pc
136
138
FTB
Pt B
Pt C
Fφ
(A/2) + ω
Ps
ROLLER 130 AT POINT A CONTACT
Pt A
FIC
Pt C
Pt B
FIA
134
132
122
PD

FIG. 14

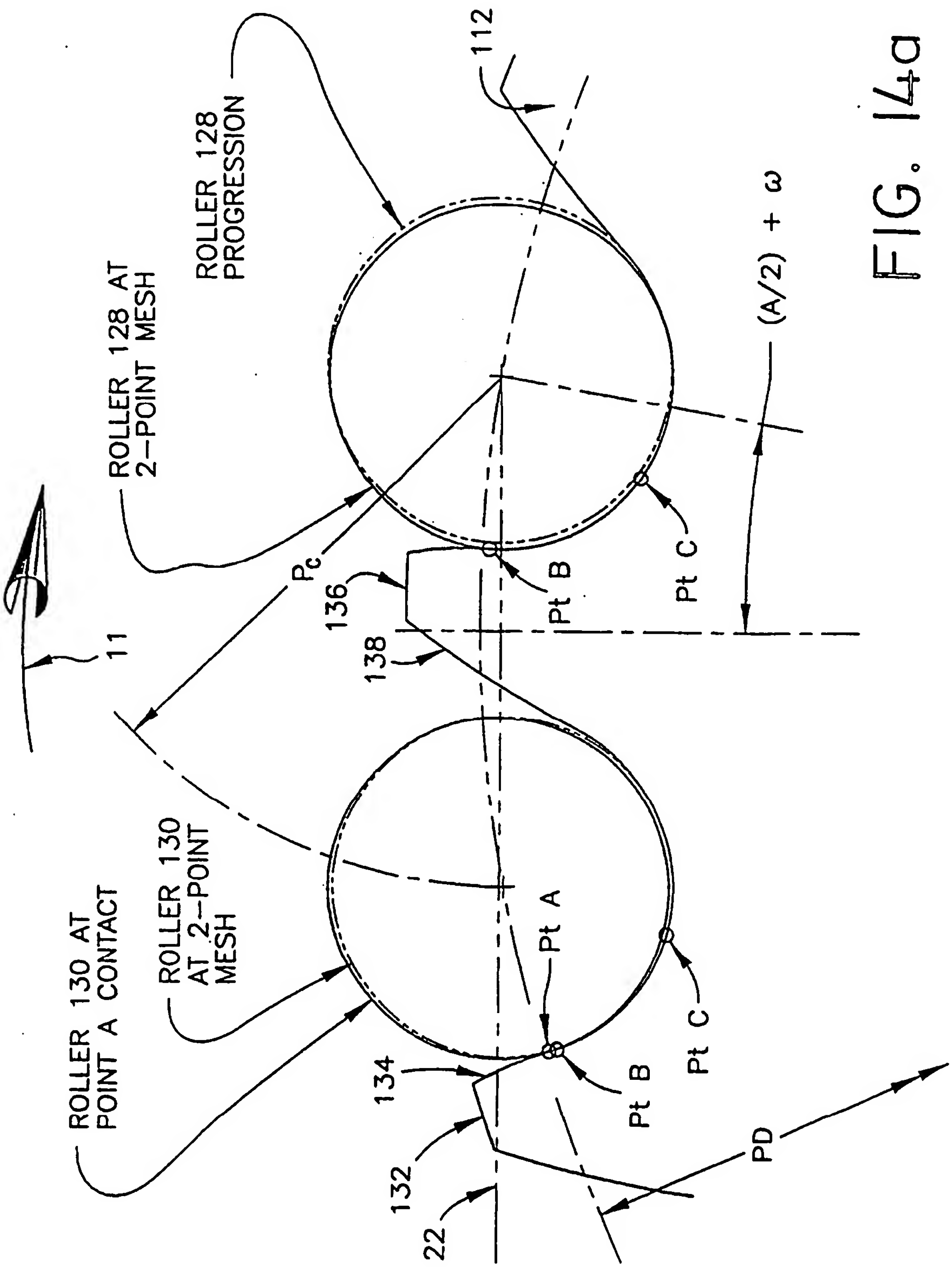

ROLLER 128 PROGRESSION
112
ROLLER 128 AT 2-POINT MESH
(A/2) + ω
11
P_c
136
Pt B
Pt C
138
ROLLER 130 AT 2-POINT MESH
ROLLER 130 AT POINT A CONTACT
Pt A
134
Pt B
Pt C
132
122
PD

FIG. 14a

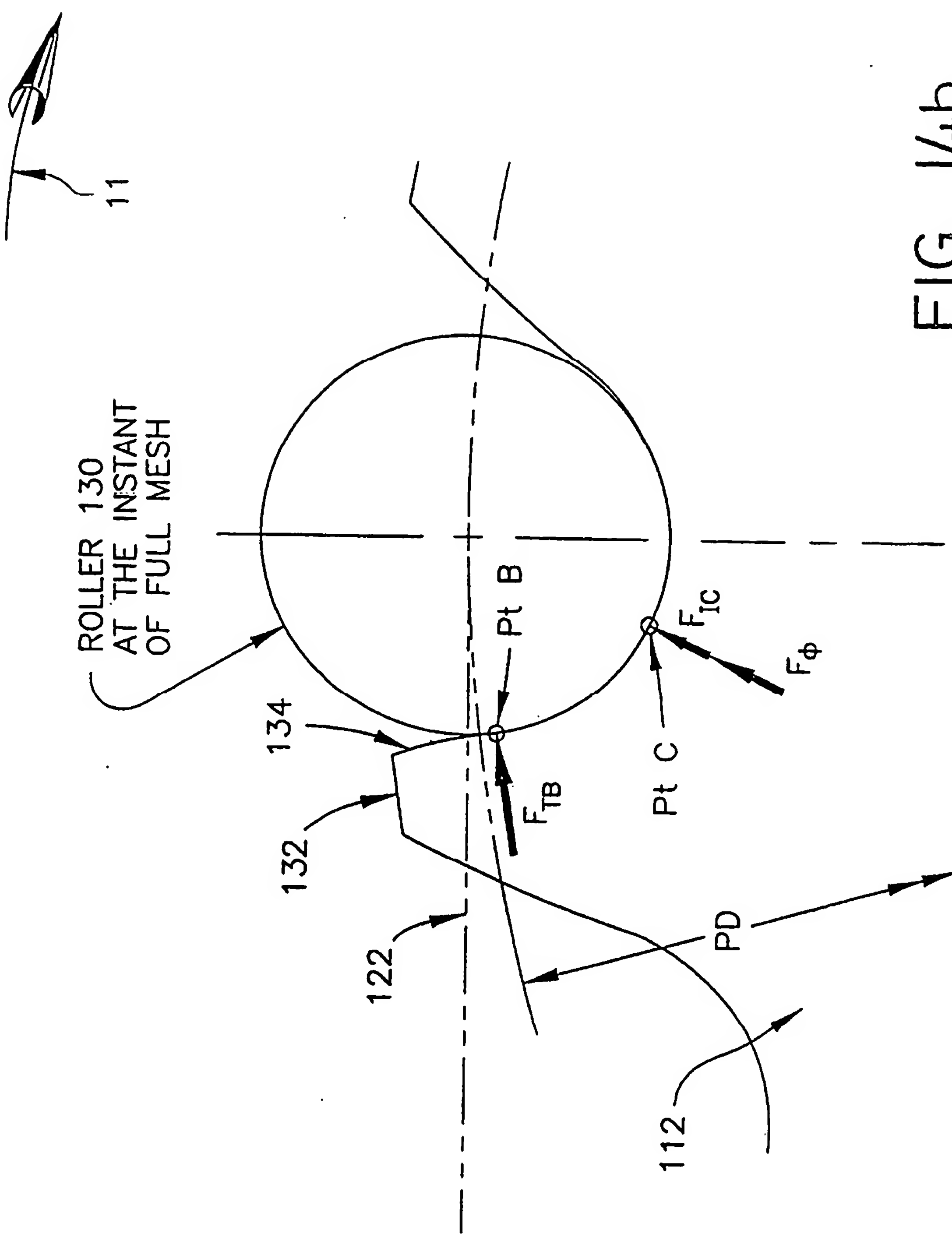

11
ROLLER 130
AT THE INSTANT
OF FULL MESH
Pt B
F_{IC}
F_{Φ}
134
132
F_{TB}
Pt C
122
PD
112

FIG. 14b

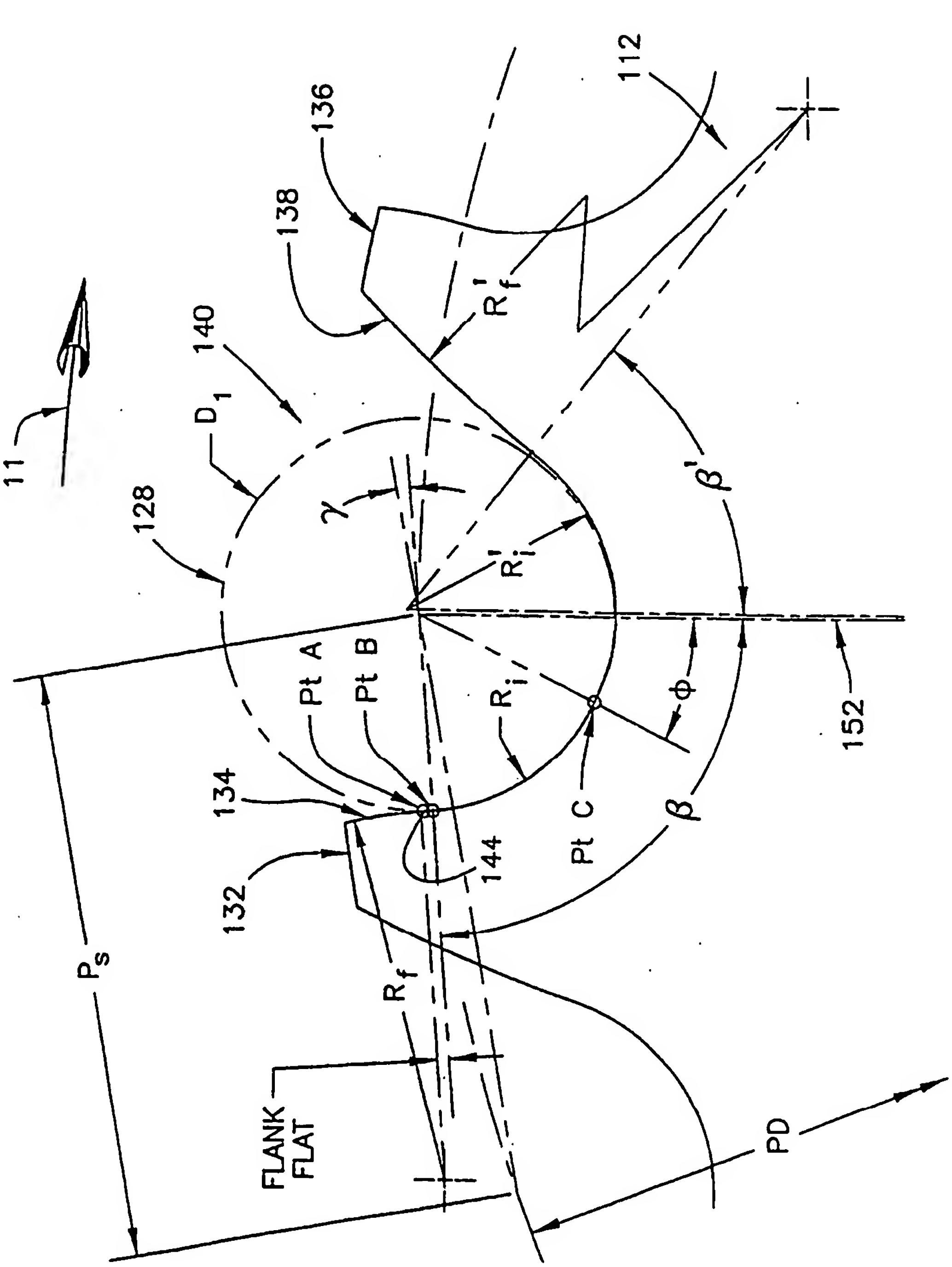

11
128
D_1
140
138
136
112
R'_f
γ
R'_i
β'
Pt A
Pt B
R_i
Pt C
ϕ
152
β
134
132
144
P_s
R_f
FLANK FLAT
PD

FIG. 15

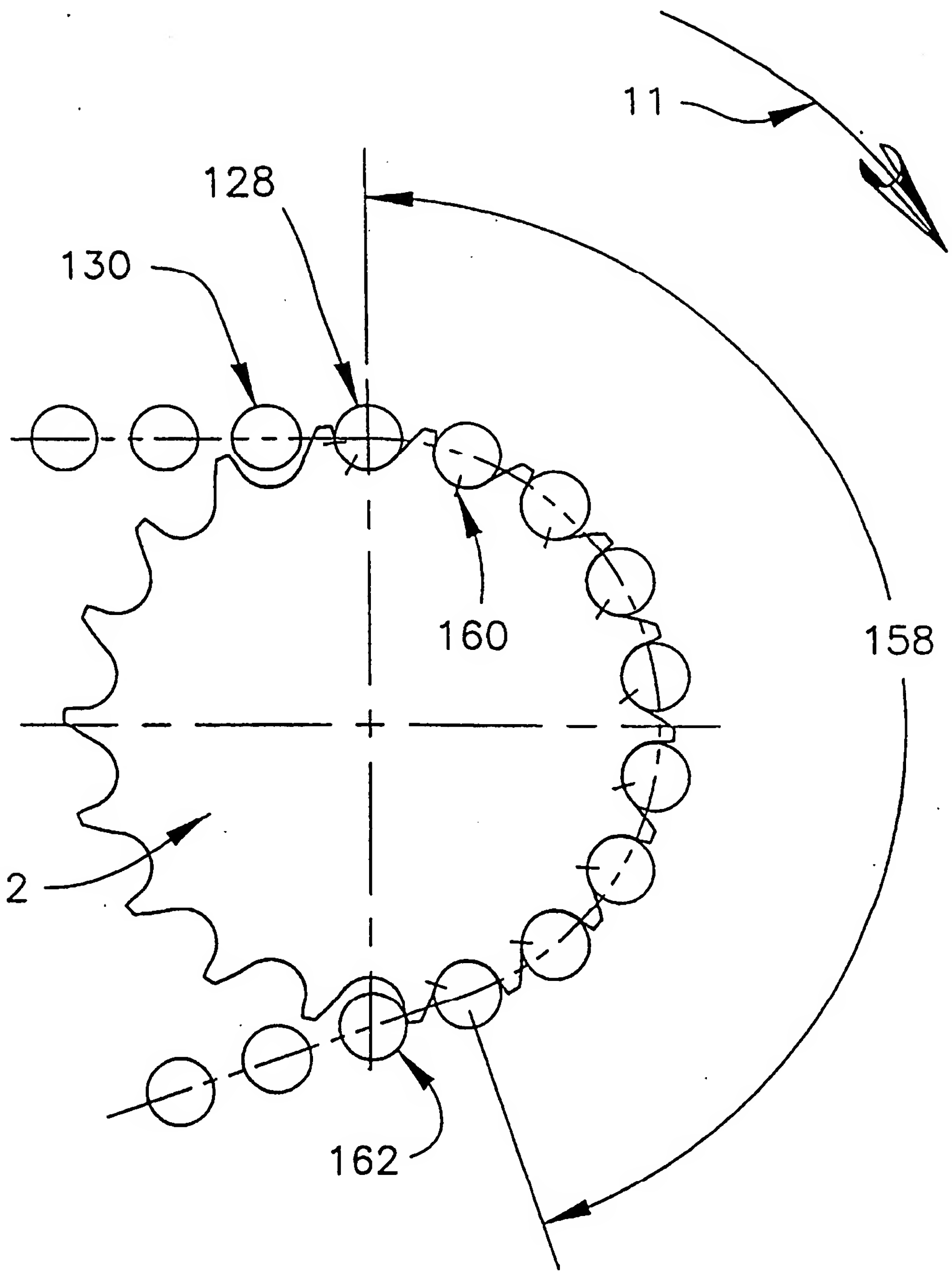


FIG. 16

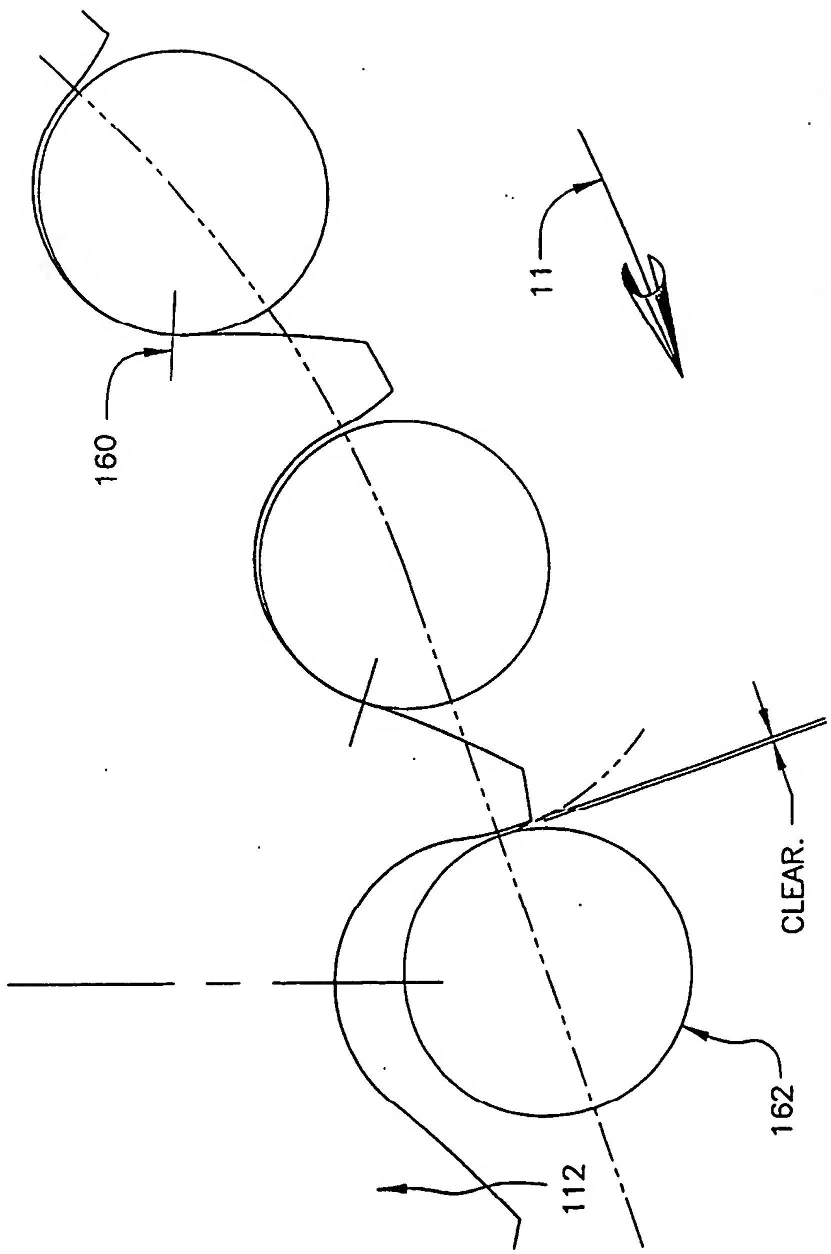


FIG. 17

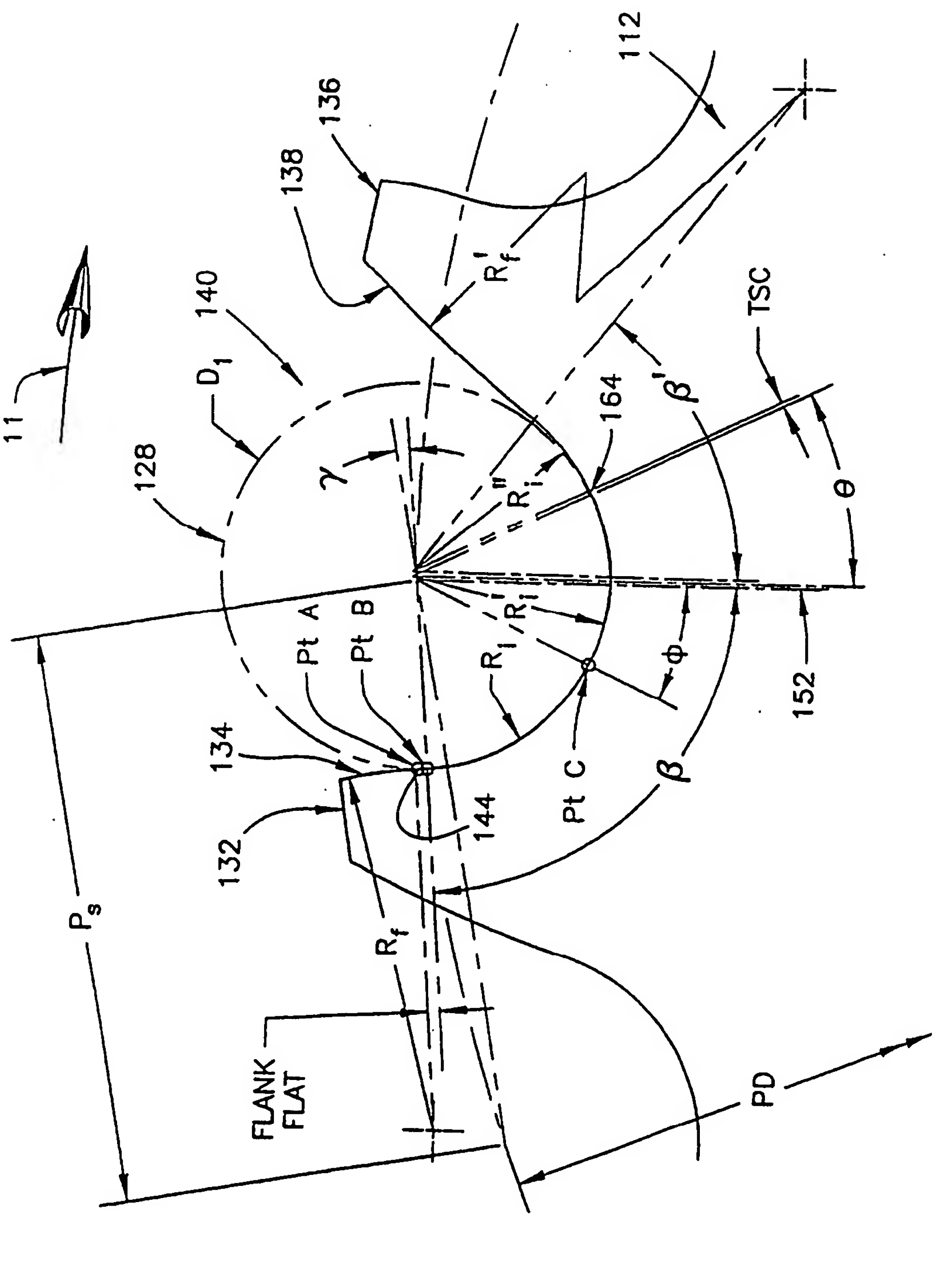

11
128
D1
140
138
136
112
γ
R'f
R''1
164
β'
TSC
θ
R1
152
φ
Pt A
Pt B
134
132
144
Pt C
β
Rf
Ps
FLANK FLAT
PD

FIG. 18

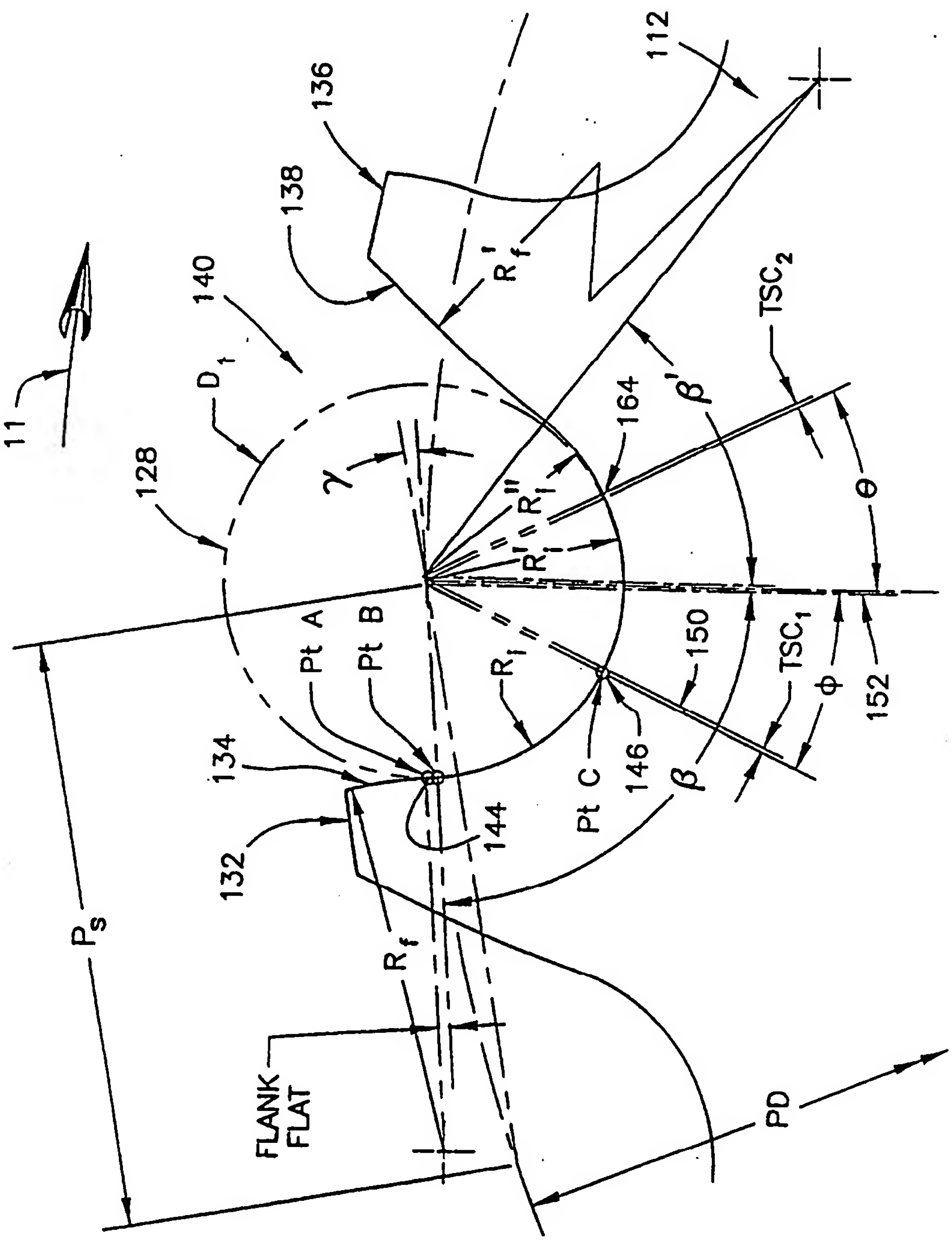

11
112
136
138
140
128
D_1
γ
R'_f
R''_1
R'_1
164
β'
TSC_2
θ
Pt A
Pt B
R_1
150
TSC_1
ϕ
152
134
132
144
Pt C
146
β
P_s
R_f
FLANK FLAT
PD

FIG. 19

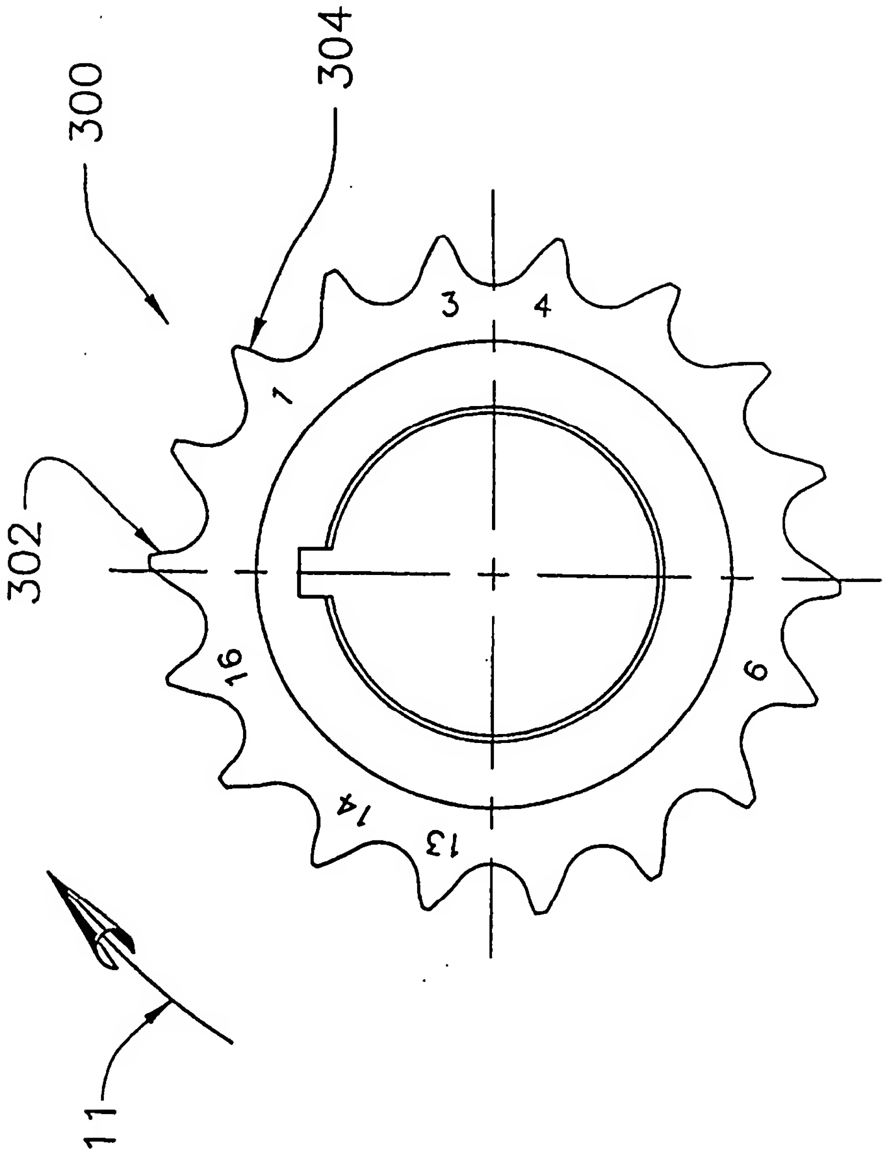
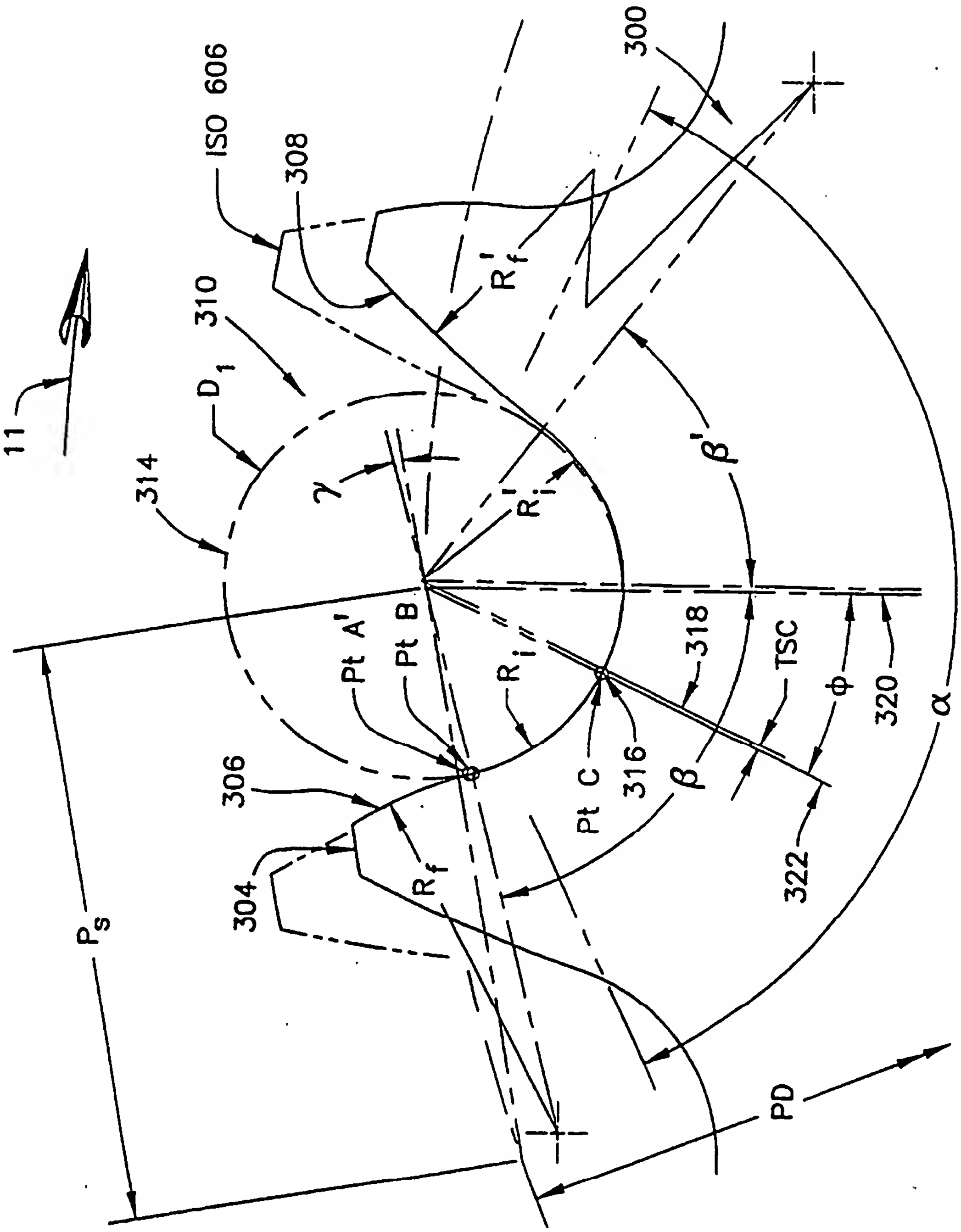


FIG. 20



ISO 606
308
300
310
D_1
11
314
R_f'
R_i'
γ
β'
Pt A'
Pt B
R_i
318
TSC
ϕ
320
α
306
Pt C
316
β
322
304
R_f
P_s
PD

FIG. 21

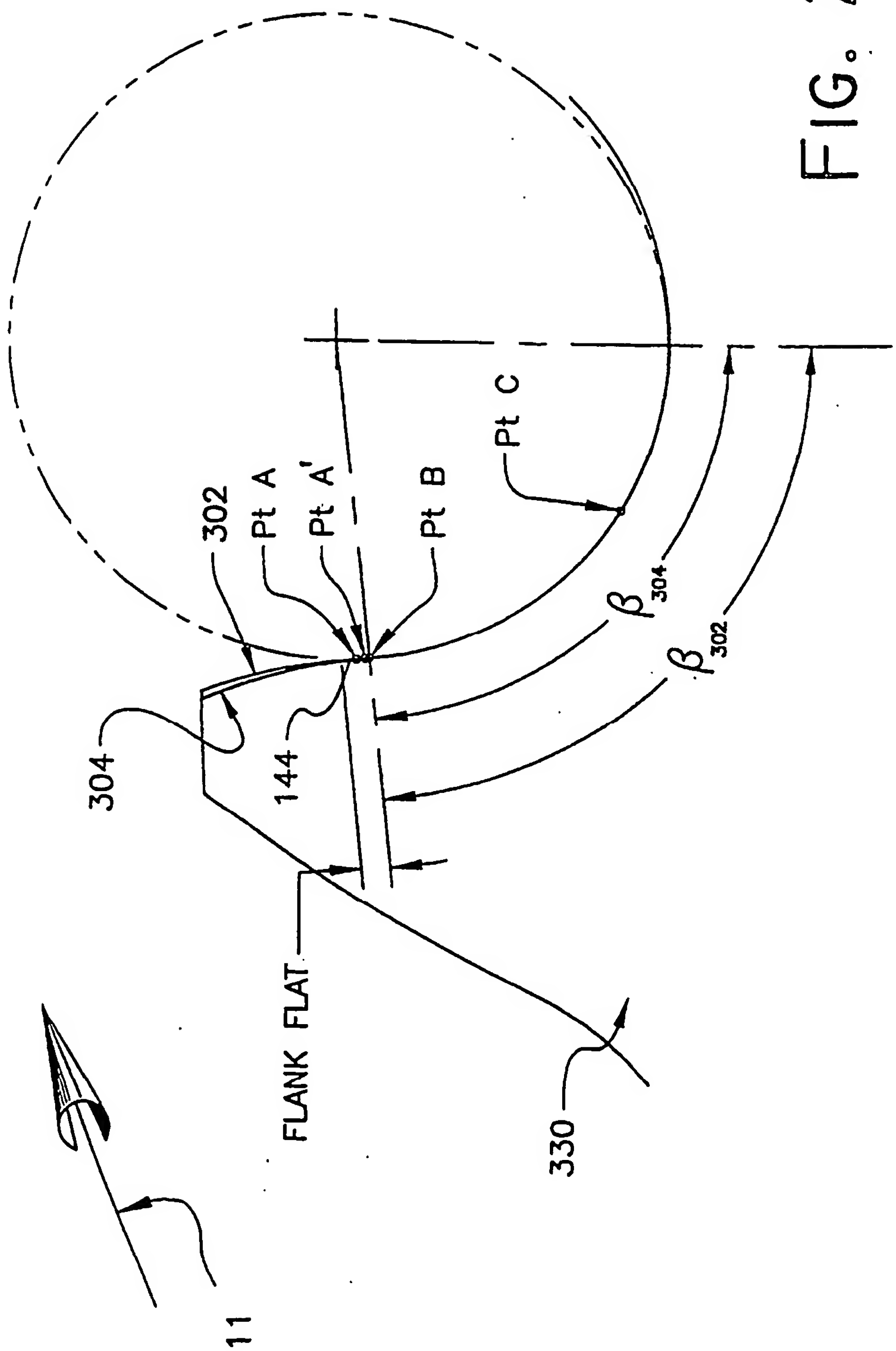


FIG. 22

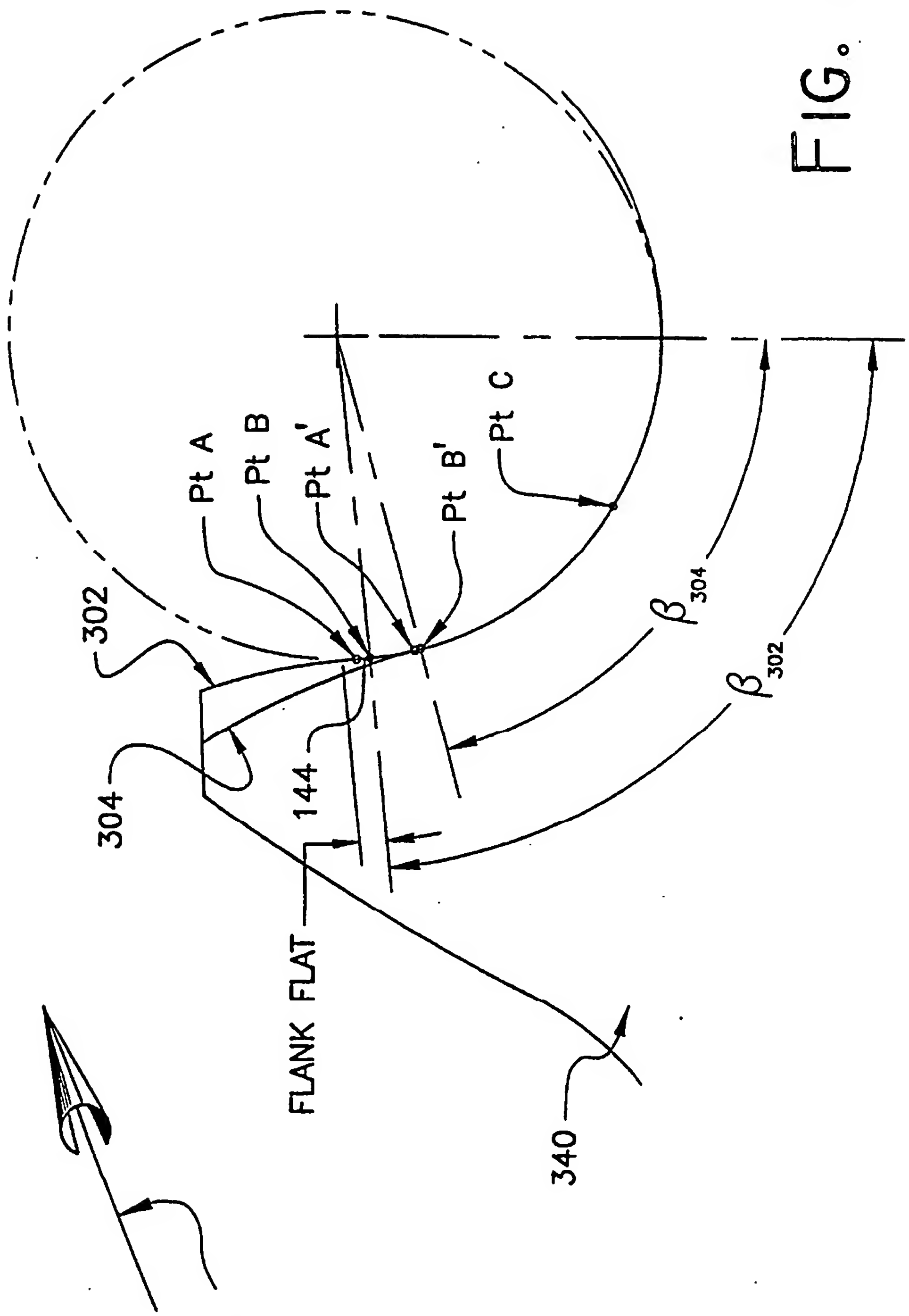

Pt A
Pt B
Pt A'
Pt B'
Pt C
302
304
144
FLANK FLAT
340
11
β_{304}
β_{302}

FIG. 23

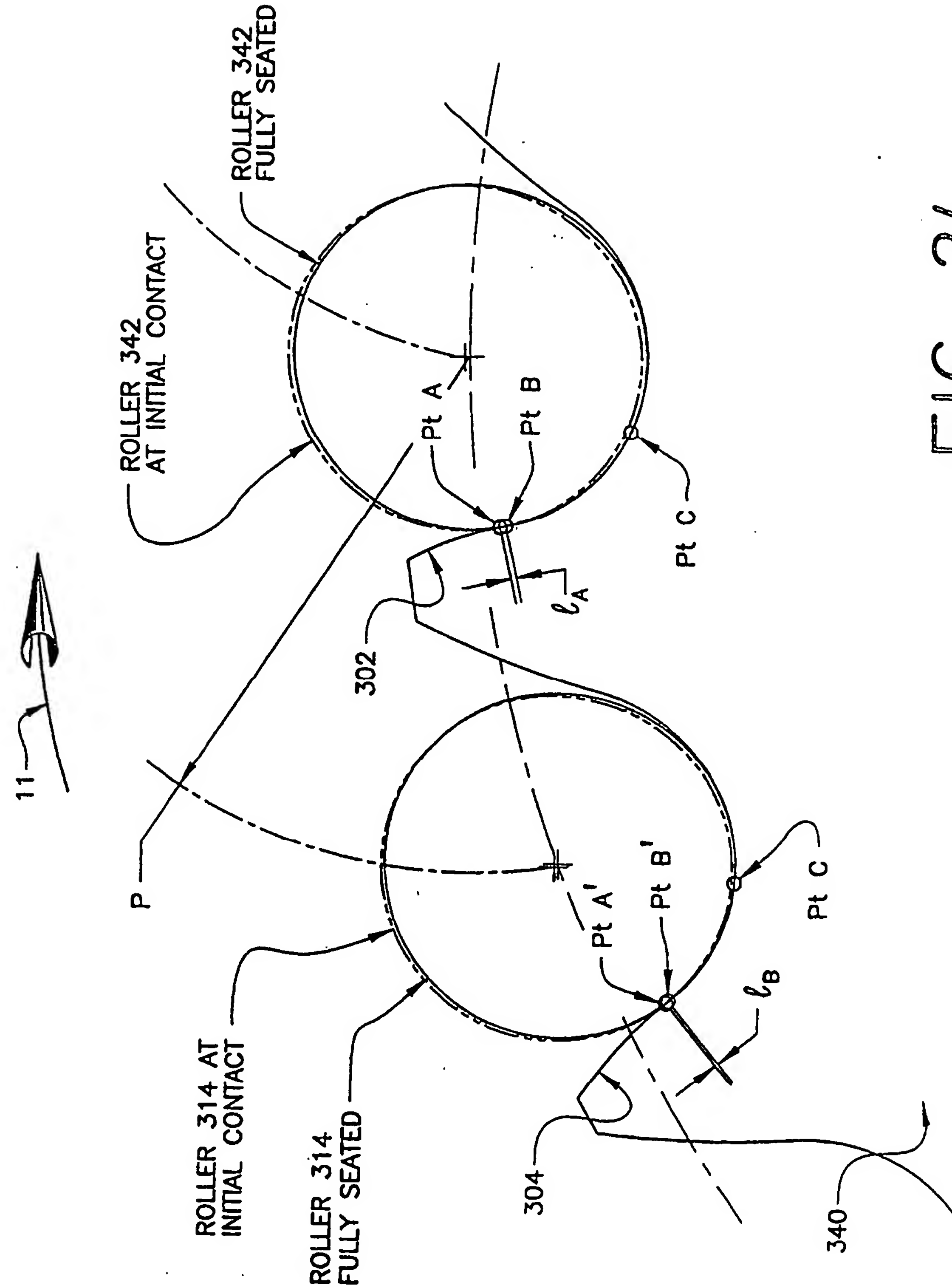

ROLLER 342 FULLY SEATED
ROLLER 342 AT INITIAL CONTACT
Pt A
Pt B
Pt C
l_A
302
11
P
ROLLER 314 AT INITIAL CONTACT
ROLLER 314 FULLY SEATED
Pt A'
Pt B'
Pt C
l_B
304
340

FIG. 24

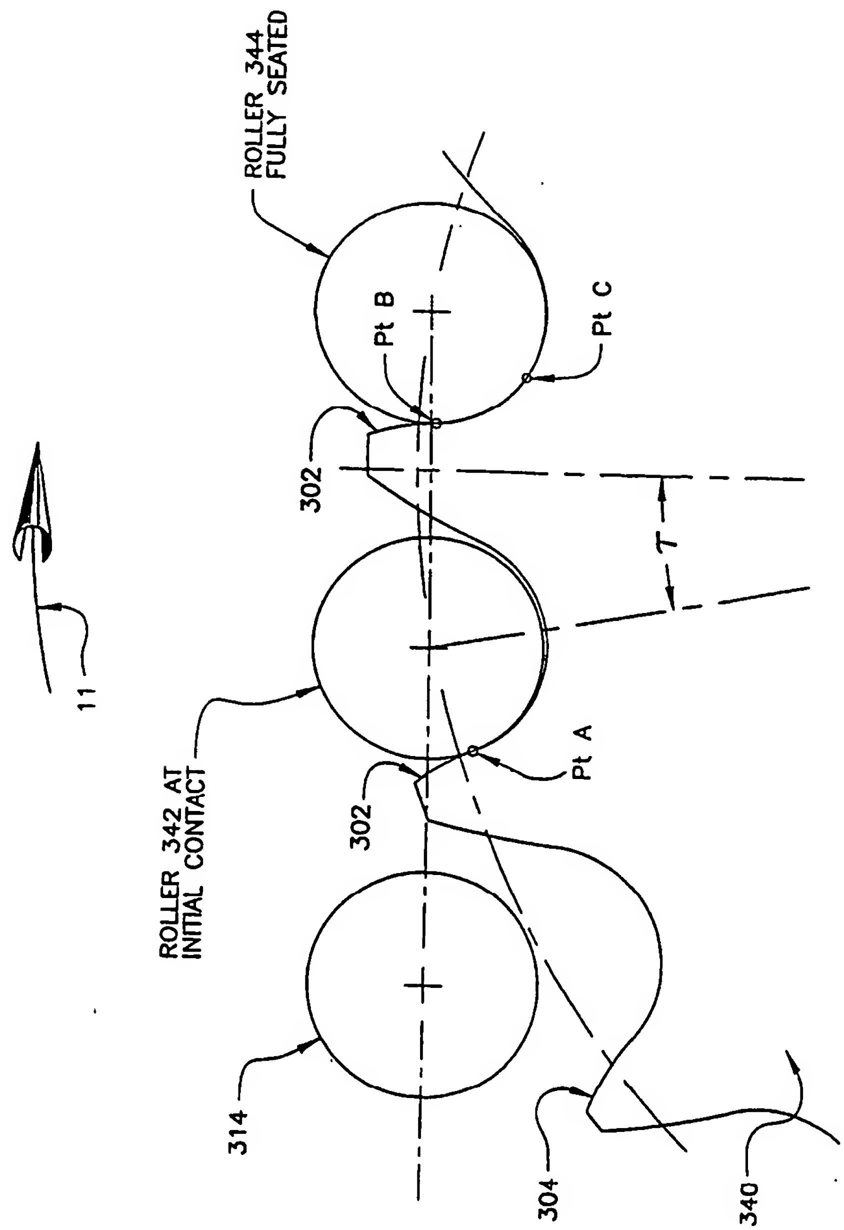

ROLLER 344
FULLY SEATED
Pt B
Pt C
302
T
11
ROLLER 342 AT
INITIAL CONTACT
Pt A
302
314
304
340

FIG. 25

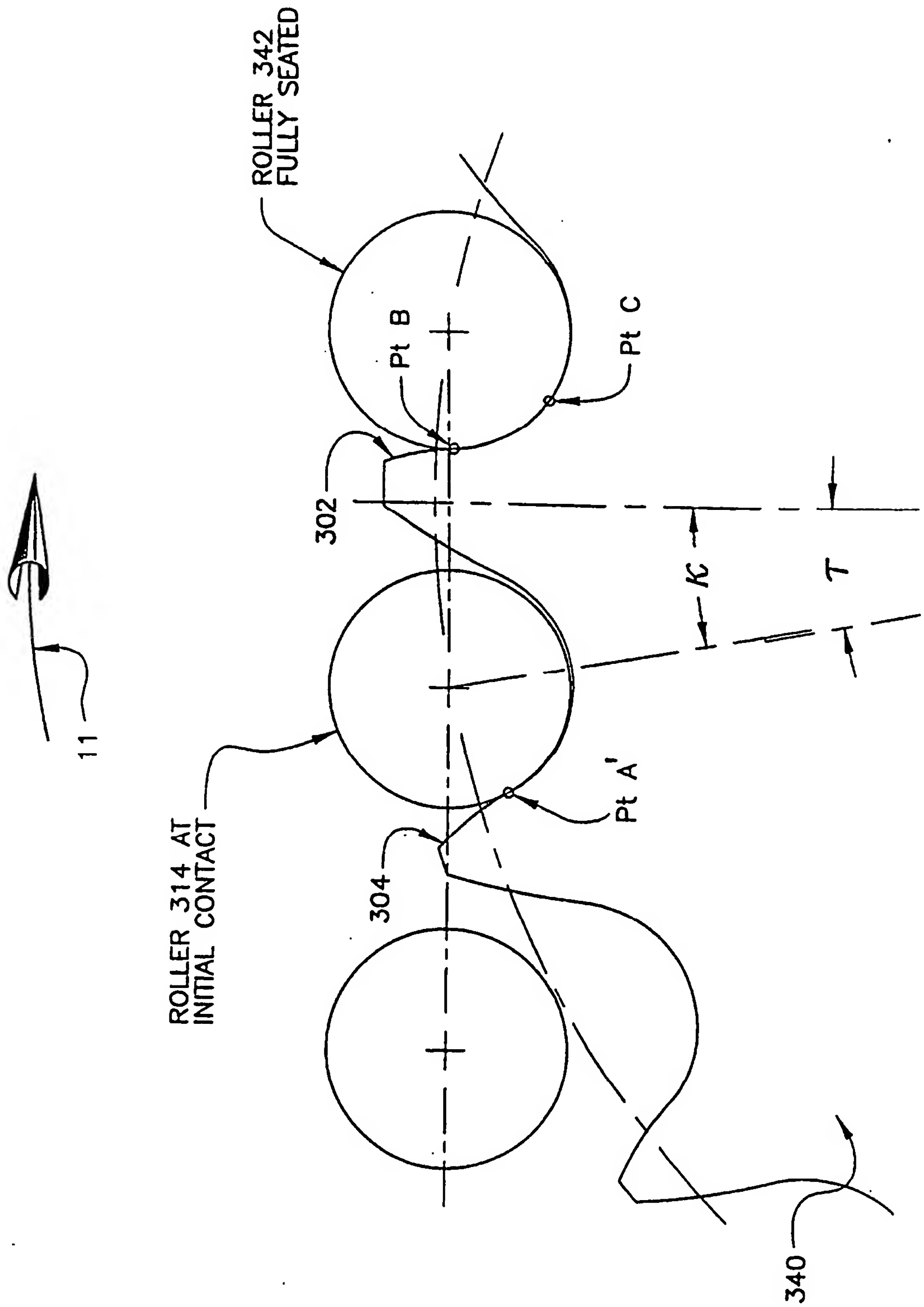

ROLLER 342
FULLY SEATED
Pt B
Pt C
302
K
T
11
ROLLER 314 AT
INITIAL CONTACT
Pt A'
304
340

FIG. 26

ROLLER CHAIN SPROCKETS

ROLLER SEATING & PRESSURE ANGLES

ISO

NO. OF TEETH Z	TOOTH ANGLE A	ROLLER SEATING ANGLE *ALPHA*		PRESSURE ANGLE *GAMMA*	
		MINIMUM	MAXIMUM	MAXIMUM	MINIMUM
18	20.000	115.00	135.00	22.500	12.500
19	18.947	115.26	135.26	22.895	12.895
20	18.000	115.50	135.50	23.250	13.250
21	17.143	115.71	135.71	23.571	13.571
22	16.364	115.91	135.91	23.864	13.864
23	15.652	116.09	136.09	24.130	14.130
24	15.000	116.25	136.25	24.375	14.375
25	14.400	116.40	136.40	24.600	14.600
26	13.846	116.54	136.54	24.808	14.808
27	13.333	116.67	136.67	25.000	15.000
28	12.857	116.79	136.79	25.179	15.179
29	12.414	116.90	136.90	25.345	15.345
30	12.000	117.00	137.00	25.500	15.500

FIG. 27
PRIOR ART

ROLLER CHAIN SPROCKETS

ROLLER SEATING & PRESSURE ANGLES

	ISO				ASYMMETRICAL					
					PROFILE 1		PROFILE 2		PROFILE 3	
NO. OF TEETH Z	ROLLER SEATING ANGLE ALPHA (MIN)	ROLLER SEATING ANGLE ALPHA (MAX)	ALPHA (MAX) /2	PRESSURE ANGLE GAMMA (MIN)	BETA (MAX)	PRESSURE ANGLE GAMMA (MIN)	BETA (MAX)	PRESSURE ANGLE GAMMA (MIN)	BETA (MAX)	PRESSURE ANGLE GAMMA (MIN)
18	115.00	135.00	67.50	12.500	73.75	6.25	80.00	0	82.00	-2
19	115.26	135.26	67.63	12.895	74.08	6.45	80.53	0	82.53	-2
20	115.50	135.50	67.75	13.250	74.38	6.63	81.00	0	83.00	-2
21	115.71	135.71	67.86	13.571	74.64	6.79	81.43	0	83.43	-2
22	115.91	135.91	67.95	13.864	74.89	6.93	81.82	0	83.82	-2
23	116.09	136.09	68.04	14.130	75.11	7.07	82.17	0	84.17	-2
24	116.25	136.25	68.13	14.375	75.31	7.19	82.50	0	84.50	-2
25	116.40	136.40	68.20	14.600	75.50	7.30	82.80	0	84.80	-2
26	116.54	136.54	68.27	14.808	75.67	7.40	83.08	0	85.08	-2
27	116.67	136.67	68.33	15.000	75.83	7.50	83.33	0	85.33	-2
28	116.79	136.79	68.39	15.179	75.98	7.59	83.57	0	85.57	-2
29	116.80	136.90	68.45	15.345	76.12	7.67	83.79	0	85.79	-2
30	117.00	137.00	68.50	15.500	76.25	7.75	84.00	0	86.00	-2

FIG. 28

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US97/13076

A. CLASSIFICATION OF SUBJECT MATTER
IPC(6) :F16H 7/06
US CL : 474/156, 900
According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
U.S. : 474/152, 153, 155, 156, 900

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
NONE

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
APS

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 4,016,772 A (CLEMENS ET AL) 12 APRIL 1977 (12/04/77) see entire document	1-25
A	US 4,116,081 A (LUTTRELL ET AL) 26 SEPTEMBER 1978 (26/09/78) see entire document	1-25
A	US 5,397,278 A (SUZUKI ET AL) 14 MARCH 1995 (14/04/95) see entire document	1-25

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
16 NOVEMBER 1997	26 JAN 1998

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